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Heating · Piping and Air Conditioning

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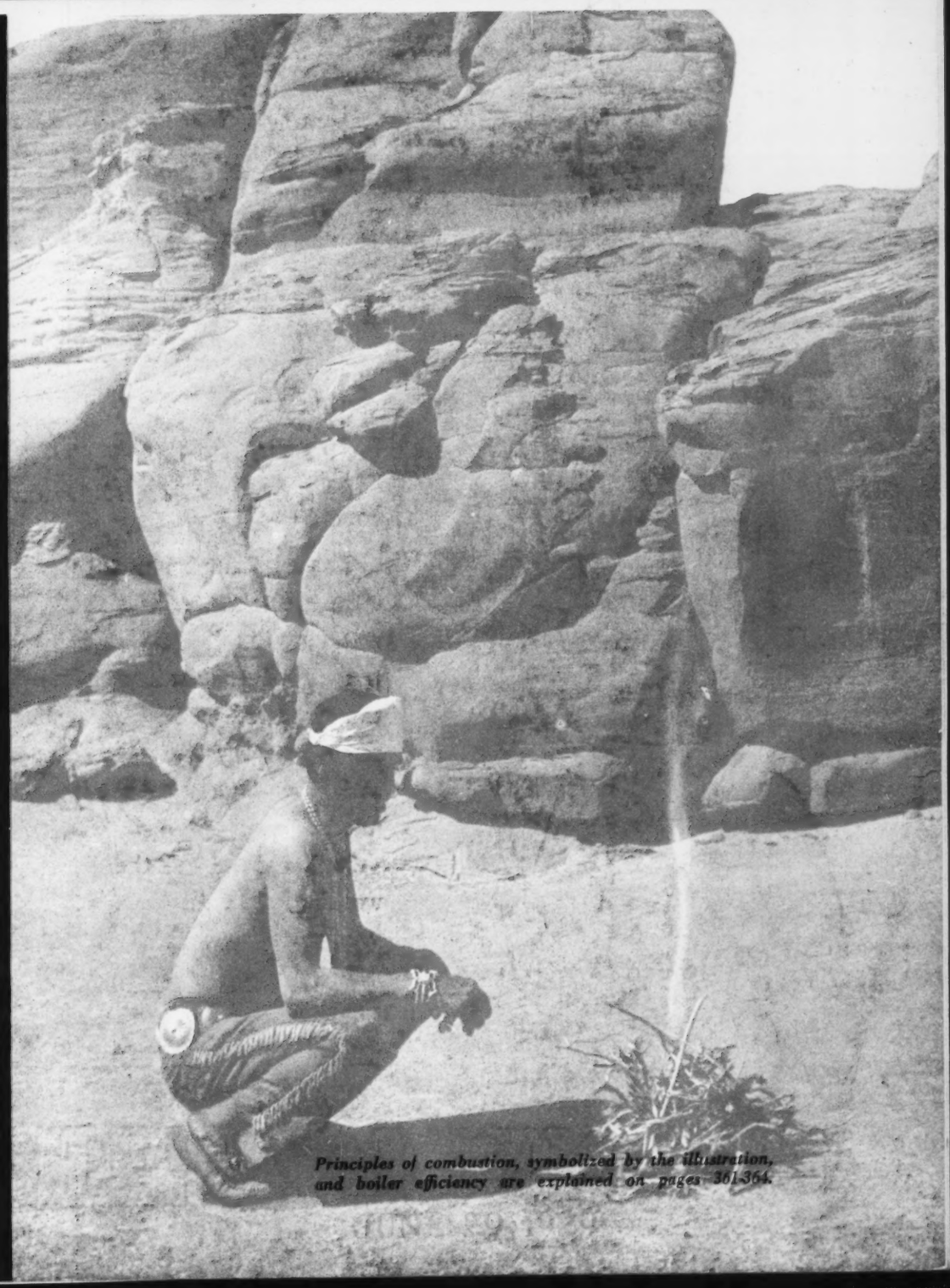

Principles of combustion, symbolized by the illustration, and boiler efficiency are explained on pages 361-364.

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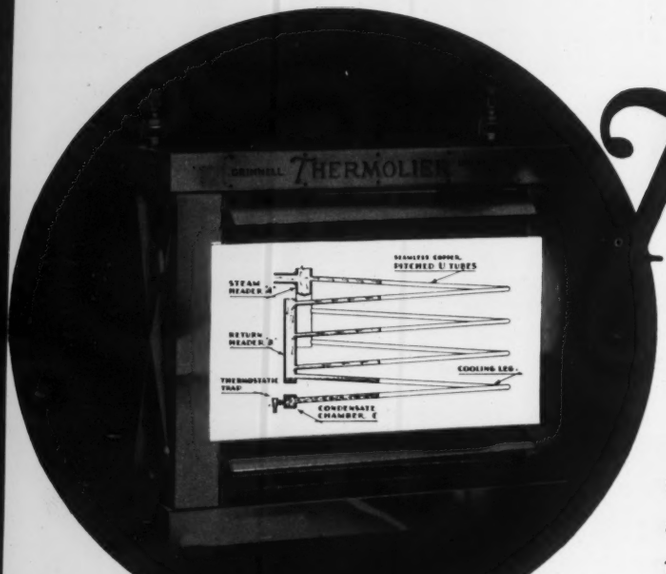

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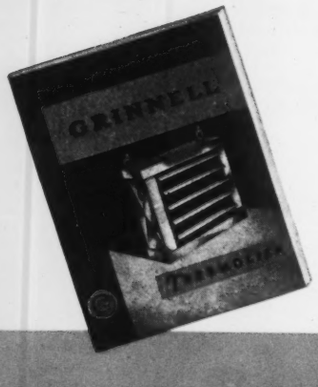

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THE EDITOR'S PAGE • July, 1936

The Editor,
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The following article in the July, 1936, H. P. & A. C. most interests me.....

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■ The editors take this opportunity to thank those readers who have returned the coupon appearing on this page in recent issues. Long distance honors go to C. G. Downie of Cape Town, South Africa, engineering assistant to the city electrical engineer; Mr. Downie voted for the Goodman article on new uses for the psychrometric chart on the April coupon, which reached us last week. . . Which article in this month's HEATING, PIPING AND AIR CONDITIONING most interests *you?* On what subjects would you like to see more articles? We'd like to know, as it will be helpful in developing material for future publication, will aid us in editing H. P. & A. C. to meet your every-day needs and interests. Your cooperation in filling in and returning the coupon on this page will be appreciated.

■ *New headquarters of the National District Heating Association are in the Engineers' Club Building, 1317 Spruce St., Philadelphia, Pa., and Wm. H. Sanford has been appointed secretary-treasurer of the Association, it was announced at the N.D.H.A.'s annual convention at Grand Rapids last month. . . . The members at the meeting had so many things to talk about that breakfast sessions supplemented the scheduled program. . . . Registration totalled over 150, included A. C. Rogers, one of the founders and first president of the N.D.H.A. . . . New officers elected are M. D. Engle, president; T. E. Purcell, first vice-president; J. R. McCausland, second vice-president; and G. D. Winans, third vice-president. Edward Lenz, F. L. Witsell, and Robert L. Fitzgerald comprise the executive committee.*

■ Anniversaries—This month will see the celebration of the centennial of the present patent system of the United States, the "Industrial Bulletin" of Arthur D. Little, Inc., notes. The Act of 1836, passed on July 4 of that year, inaugurated a regular system of examination of each application before the grant of a patent. . . . First patent under this Act was issued July 13, 1836, to John Ruggles of Thomaston, Maine, and is No. 1 of the present series. . . . Current numbers are over 2,040,000. . . . Julien P. Friez & Sons, Inc., of Baltimore, Maryland, celebrated its 60th year of business last month. The company was founded in 1876 by Julien P. Friez, father of the present president, Lucien L. Friez. . . . Schenectady, New York, marked its half century of electrical progress last month, commemorating the establishment of the electrical industry there by Thomas Alva Edison fifty years ago. . . . Among the speakers were Owen D. Young, Dr. George R. Lunn, W. S. Barstow, and Charles A. Edison, son of the famous inventor, who identified himself as "one of Mr. Edison's later experiments." . . . From the Edison Machine Works grew the Schenectady Works of the General Electric Co.

■ *The University of Illinois department of mechanical engineering is installing a complete central station type air conditioning plant to be used as laboratory equipment by undergraduates in the regular heating, ventilating and air conditioning courses and also to be available for research. . . . The compressor is to be a 14 ton "Freon" machine and the fan will have a capacity of 3,000 cfm. . . . The plant is designed to enable many types of operation—air washer, direct expansion, etc.—glass panels will be provided for observing operation, complete accessory equipment and controls will be incorporated. . . . Air conditioning students need and want experience with actual equipment, the University feels.*

■ It's difficult to imagine a more welcome application of comfort air conditioning than the convention meeting room. Long reports demand close attention; chairs get harder and harder toward the end of a lengthy session; most of those attending a convention stay up too late the night before, for one reason or another. . . . When hot, humid weather adds its discomforts, "convention efficiency" is bound to drop. . . . A number of hotels now have air conditioned meeting rooms; our personal feeling is that we won't step into another one that isn't. . . . Note: These remarks do not refer to any specific meeting held recently, but rather to a composite of memories since we've been convention-goers.

■ *The Air Conditioning Manufacturers Association has adopted a standard form of warranty which is recommended for the use of members. . . . The Division of Simplified Practice of the National Bureau of Standards, Washington, D. C., has mailed copies of a proposed revision of Simplified Practice Recommendation R175, Steel Horizontal Firebox Heating Boilers, to interested parties for consideration and approval; it covers a simplified schedule of standard ratings and pipe connection openings. . . . Copies are available without charge from the Division.*

■ R. C. Goddard, vice-president of The Steel Products Engrg. Co. in charge of the Combustioneer Division, was elected president of the National Association of Stoker Manufacturers at their convention last month. . . . Ezra Frick, Frick Co. head, was elected an honorary member of the Refrigerating Machinery Association at its recent spring meeting. He has been active in steam engineering and refrigeration since the early '70s, is a charter member and former president of the A.S.R.E. . . . While a lad of seven, he saw General Lee's troops pass through Waynesboro at the time of the battle of Gettysburg. . . . The Association of American Railroads announced last month an air conditioning research program "to determine to what extent improvements, if any, can be made in the systems now in use and how far they can be standardized so as to reduce costs of installation and maintenance"; director of the project is L. W. Wallace. . . . Selected railroad men are to be given instruction in air conditioning at Ohio State University under the direction of Prof. A. I. Brown. . . . According to the Association, there are nearly 7,000 air conditioned cars in operation. . . . Air conditioning is playing a prominent role at the Texas Centennial Exposition; a feature article about the Fair is being written by a Texas engineer for publication in H. P. & A. C.

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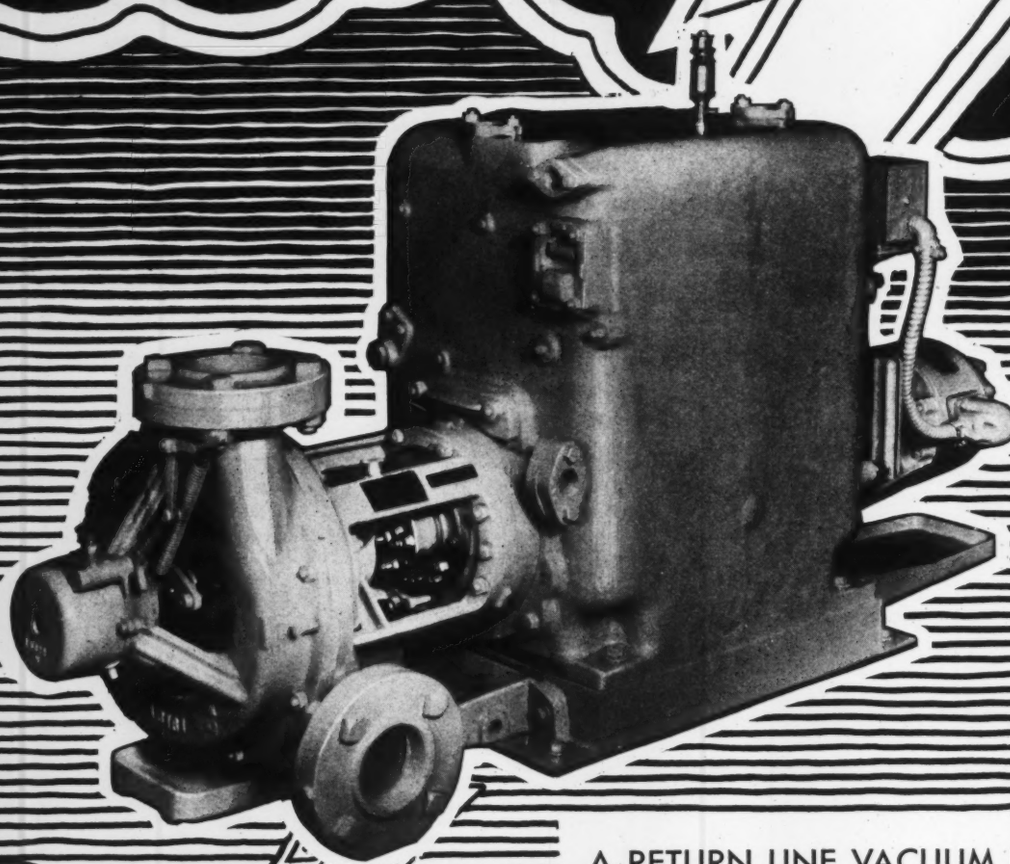

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Heating · Piping and Air Conditioning

Vol. 8, No. 7 • *July, 1936*

What the Operating Engineer Should Know

The Combustion Process and Boiler Efficiency

By Paul R. Unger* *and* Kalman Steiner†

COMBUSTION efficiency is a subject on which much has been said and written, but it will undoubtedly be worthwhile for most operating men briefly to review the fundamental concepts. Combustion is a chemical process, and its regulation is a chemical problem, so that for those not trained in chemistry it is easy to form mis-conceptions of the process and to misunderstand the terms used.

Only two of the 92 elements known to science are encountered as combustible matter in fuels—whether the fuel be coal, oil, gas, wood or any other substance. These two elements are carbon and hydrogen; a third element, sulphur, if present, will burn with hydrogen and carbon but it is not necessary to consider sulphur as a source of heat as it occurs only in very small percentages in fuels. However, in order to burn a fuel it is necessary to supply oxygen to it, and as the ordinary source of oxygen for this purpose is air, which also contains nitrogen, we have two more elements to consider. So it is possible to restrict a discussion of the chemistry of combustion to a consideration of but four elements—carbon, hydrogen, oxygen and nitrogen—with sulphur as a possible fifth.

Efficient operation of a boiler plant requires a knowledge of the process of combustion and the meaning of a flue gas analysis. For those not trained in chemistry, it is easy to form mis-conceptions and to misunderstand the terms commonly used. The authors have therefore considered an assumed volume of air as it successively enters the firebox, participates in the combustion process, and emerges as flue gas, showing in a very clear way where the losses occur

By investigating the behavior of these elements it is possible to establish the laws of combustion, and from a knowledge of those laws, it is in turn possible to regulate combustion so that the highest possible percentage of available heat is utilized. But the chemical changes occurring in a firebox are so rapid that it is difficult to observe the elements while undergoing combustion. Instead, the simpler procedure is adopted of taking samples of and investigating the substances produced by combustion—the so-called flue gases. Most operating control is therefore based upon analysis of the flue gases and interpretation of the analysis.

Combustion Process Illustrated

To illustrate, it will serve the purpose very nicely to follow an hypothetical volume of air as it successively enters the firebox, participates in combustion, and then emerges as flue gas. The volume used is the *mol,* which is 359 cu ft at 32 F and 29.9 in. barometric pressure. Chemists use the mol as the unit of volume because the weight in pounds of a mol of *any* gas equals numerically the molecular weight of the gas. It has been shown by chemists that all gases contain the same number of molecules per mol.

For all practical purposes, air may be considered here as consisting of 20.9 per cent oxygen (O_2) and 79.1 per cent nitrogen (N_2) by volume. Fig. 1 shows an hypothetical mol of air, except that while of course the air is a uniform and homogeneous mixture of gases, for the present discussion it is shown as if the two component gases were separated.

Now if such a thing were possible, and this mol of air were to undergo changes in *complete and perfect* combustion, and the fuel used were pure carbon (C), it

*Mid-West Heat Service Co., Chicago, Ill.
†Ace Engineering Co., Chicago, Ill.
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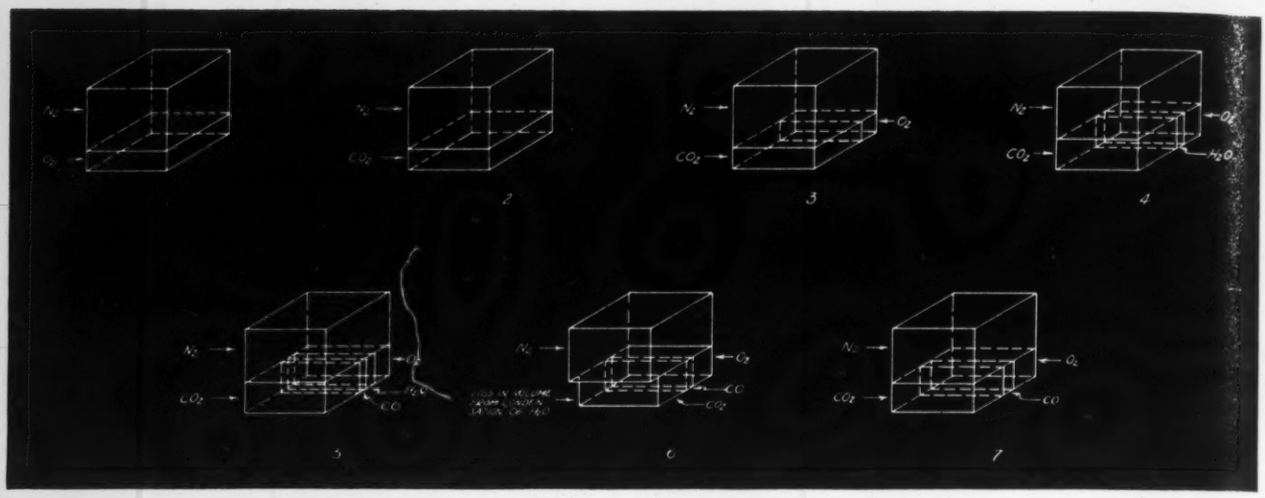

Figs. 1-7—An assumed volume of air successively entering the firebox, participating in combustion and emerging as flue gas, illustrating the process explained in the text

would appear as flue gas with a composition as shown in Fig. 2. The nitrogen would be unchanged; the oxygen would be quantitatively changed into carbon dioxide (CO_2). But since perfect combustion cannot ordinarily occur, there would actually be some unused oxygen remaining in the flue gas, so that it would really appear as shown in Fig. 3 (still assuming that the fuel is *pure* carbon). As its temperature would undoubtedly be higher, the mol would occupy a greater space than when at room temperature.

All commercial fuels, however, contain hydrogen in varying amounts. This hydrogen combines with some of the oxygen to form water (H_2O), which at the temperature of the combustion chamber or of the chimney passages if above 212 F, is also a gas. Hence, the flue gas from our mol of air is now as shown in Fig. 4.

It is important to note at this point that each molecule of oxygen (O_2) forms two molecules of water (H_2O) so that the combustion of hydrogen from the fuel causes the total number of molecules in the flue gas to become greater than the original number of molecules of the air which has been used. This same phenomenon occurs when carbon monoxide is formed. Hence, the volume occupied by the flue gas illustrated in Figs. 4 and 5 is greater than the volume occupied by the mol of air which was the origin (aside of course from expansion which results from temperature rise, and which affects all gases equally). As a result, the 0.791 mol of nitrogen which entered with the air and which remains unchanged in amount although expanded by the rise in temperature is now actually a smaller percentage of the total.

Hydrogen always burns to form water, there being no other possibility from the combustion of this element. In the case of carbon, however, there are two possibilities; in addition to carbon dioxide, carbon may form an intermediate product, carbon monoxide (CO) which is also a gas and which contains only one atom of oxygen per molecule. When CO is present in the flue gas (and the combustion is therefore incomplete), the transformed mol must be pictured as in Fig. 5.

Of course, water—or rather steam—can thus be pictured in the flue gas only so long as its temperature remains above 212 F. While this temperature is usually exceeded in the average breeching or chimney, the flue gas is brought down to room temperature when sampled and introduced into an Orsat apparatus or other analyzing instrument. Immediately that this occurs, the water condenses and the volume of the products of combustion decreases, aside of course from that due to the contraction of the remaining (permanent) gases due to reduction in temperature. At room temperature therefore, the gases appear as in Fig. 6.

In Fig. 7 is shown a more logical representation of these residual gases. The important difference now is that because of the condensation of water vapor each constituent gas represents a greater percentage than when water vapor was a component of the gas mixture. For instance, suppose the composition, before condensation, to have been: N_2, 75.3 per cent; CO_2, 11.3 per cent; O_2, 3.7 per cent; CO, 0.3 per cent; and H_2O, 9.4 per cent. As soon as the water condenses, the gases which previously totalled only 90.6 per cent of the mixture now become 100 per cent. Accordingly, the new percentage composition is obtained by dividing each remaining constituent of the previous mixture by 0.906 to get the corresponding percentage of each constituent in the new mixture. Thus, nitrogen is now present to the extent of $75.3/0.906 = 83.1$ per cent. Similarly, the percentages of the other gases are, roughly, carbon dioxide, 12.5; oxygen, 4.1; and carbon monoxide, 0.3+.

It will soon be seen that the apparent percentage of nitrogen in flue gas as indicated by an Orsat apparatus furnishes a valuable means of checking other data, from the knowledge that the apparent percentage results from the 79.1 per cent of the original air.

It is now possible to draw some quantitative deductions from the flue gas composition. It is apparent that when ordinary air is used for supporting combustion, and when the fuel itself is pure carbon, and were the combustion to be complete and perfect, then the maximum possible CO_2 percentage in flue gas is obtained, and that percentage is 20.9. Under actual conditions, however, the attained percentage is lower because combustion is never perfect: that is, excess oxygen remains, combustion may not always be complete in that CO may be formed, and the burning of hydrogen to form water introduces a product

of combustion other than an oxide of carbon. Just what is the best possible or attainable CO_2 is brought out in the following paragraphs.

Use of Flue Gas Analysis

It is customary when using the complete Orsat apparatus to determine the percentage of CO_2 and O_2 in flue gas, and then to take the difference between their sum and 100 per cent as the percentage of nitrogen. Some types of gas analysis apparatus also provide an absorption tube for CO, but the determination of CO is a matter of some analytical difficulty and it is doubtful if an Orsat can give a dependable reading of this. For checking most combustion processes it is sufficient to determine carbon dioxide and oxygen and to figure nitrogen by difference.

Suppose such a case where the analytical results are: CO_2, 12.5 per cent; O_2, 4.1 per cent; and N_2, 83.4 per cent. From these results numerous important calculations can be made.

It is known that the composition of air is 20.9 per cent oxygen and 79.1 per cent nitrogen. Since 100 mols of flue gas must contain 83.4 mols of nitrogen, and this nitrogen must have come from air which contains only 79.1 mols of nitrogen per 100 mols of air, it follows that it would require 83.4/0.791, or 105.4 mols of air to give 100 mols of flue gas.

There is a total of 16.6 mols of O_2 in the 100 mols of dry flue gas—12.5 as carbon dioxide and 4.1 oxygen. But the 105.4 mols of air which formed this 100 mols of flue gas contained 105.4 × 0.209 or 22.0 mols of oxygen. The difference, 22.0 − 16.6 or 5.4, must have combined with hydrogen to form water, which of course has condensed out and does not appear in the analysis of the dry flue gas. This 5.4 mols of oxygen as water has combined with 10.8 mols of hydrogen.

It is now possible to tell something of the fuel which has been burned. The only carbon in the flue gas is the 12.5 mols occurring as the dioxide. By multiplying 12.5 mols by the *atomic* weight of carbon, which is 12, we find that 150 lb of carbon were burned per 100 mols of dry flue gas produced, and similarly by multiplying 10.8 mols of hydrogen by the *molecular* weight (2.02) it follows that 21.8 lb of hydrogen were consumed. The total weight of the fuel[1] burned was therefore 150 + 21.8 or 171.8 lb; by composition it contained 150/171.8 or 87.3 per cent carbon and 12.7 per cent hydrogen.

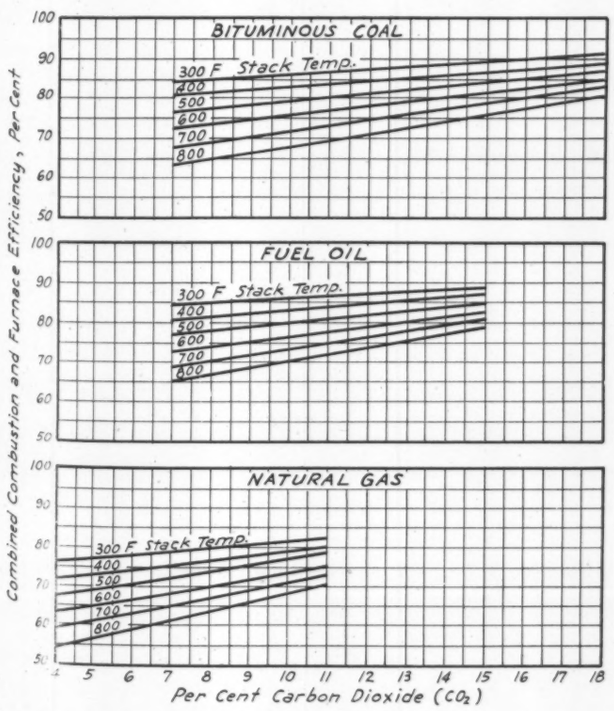

Fig. 8. Efficiency curves for boilers, indicating average stack loss for various percentages of CO_2 and various stack temperatures

Returning to the flue gas again, one of the most important deductions to be made from the analysis is the percentage of excess air. This is most conveniently made from the oxygen balance, since of course the oxygen comes from air and therefore a quantitative statement concerning either is applicable to the other. The necessary O_2 is that which has entered into combustion, and the unnecessary is that which remains uncombined in the flue gases; the ratio of the two gives the percentage of excess.

The unnecessary oxygen is shown by the flue gas analysis to be 4.1 per cent. The necessary is that which has combined with carbon and hydrogen, and for this case is 12.5 + 5.4 = 17.9 per cent. The percentage of excess oxygen, and hence of excess air, is therefore established as (4.1/17.9) × 100 = 22.9 per cent.

So far nothing has been said of the amount of heat liberated in a combustion reaction nor have the relationships of gas volume changes with respect to temperature and pressure been discussed. It now becomes necessary to apply the principles of physics known as the *gas laws*. Most technical men are familiar with the computations for correcting gas volumes for temperature and pressure. Such knowledge will be assumed here; anyone not familiar with the subject can refer to any elementary text on physics, or any engineering handbook.

It has been shown that per 100 mols of flue gas there were required 105.4 mols of air and the fuel burned was 171.8 lb. This can be converted into an expression for cu ft of air per lb of fuel by applying the gas laws. Dividing, we secure mols of air per lb of fuel, 105.4/171.8 = 0.613. Since a mol at standard conditions occupies 359 cu ft, 0.613 mol will be 220.1 cu ft at 32 F and 29.9 in.

Now suppose the temperature of the air entering the firebox to be 78 F and the barometric pressure to be 30.3 in. Then the volume of air used per lb of fuel is

$$220.1 \times \frac{460 + 78}{460 + 32} \times \frac{29.9}{30.3} = 237.4 \text{ cu ft}$$

It is also possible to calculate the volume of total flue gases produced per lb of fuel burned. For each 100 mols of dry flue gas there are also 10.8 mols of water vapor, which makes a total 110.8 mols of total flue gas to 171.8 lb of fuel, or 0.645 mol per lb. If the stack gas temperature is 490 F and the stack draft is 0.5 in., the flue gas volume per lb of fuel is

$$0.645 \times 359 \times \frac{460 + 490}{460 + 32} \times \frac{29.9}{29.4} = 456 \text{ cu ft per lb fuel}$$

There remains one more result to be figured from the flue gas analysis. If the gross calorific value of the fuel is 18,300 Btu per lb,[2] and if the heat of condensation of water is 970.4 Btu per lb, then the percentage of the gross

[1] As used in this article, the word "fuel" means "combustible." For coal, weight of combustible is weight of fuel less weight of ash.

[2] The fuel in this case being No. 6 oil.

fuel value lost through non-condensation of the water vapor formed in combustion is evaluated as follows: 10.8 mols of water vapor formed per 171.8 lb of fuel burned gives 0.063 mols per lb, and since the molal weight of water vapor is 18, the weight of water vapor formed per lb of fuel burned is 1.13 lb. The percentage of heat loss is then

$$\frac{1.13 \times 970.4}{18{,}300} \times 100 = 6.0 \text{ per cent}$$

The molal specific heat of the flue gases of this example can be shown to be 7.66, meaning that 7.66 Btu are required to raise the temperature of the flue gases 1 F. Hence, the total heat carried away in the flue gases must be, since the gases are heated from 78 to 490

$$0.645 \times (490 - 78) \times 7.66 = 2036 \text{ Btu per lb fuel}$$

Dividing this heat by the total heat content of the fuel, 2036/18,300 = 11.1 per cent, gives the per cent of heat loss as sensible heat of the flue gases.

To summarize the losses, the latent heat of formation of water vapor takes away 6 per cent and the sensible heat of the flue gases takes another 11.1 per cent, so that the total stack loss is 17.1 per cent. Were there no other losses, the efficiency of the boiler would then be 82.9 per cent. But some heat will be lost by radiation from the boiler and furnace into the floor. As these losses cannot be directly determined, only an actual evaporation test will give the real efficiency of the boiler. The difference between the real efficiency and the total remaining after deducting for water and flue loss will give the radiation and conduction losses. For instance, should the evaporation test show an actual efficiency of 79.1 per cent, then the undetermined losses by radiation and conduction would be 3.8 per cent.

Fig. 8 shows the average stack loss (water vapor and sensible heat) for various percentages of CO_2 and various stack temperatures for three different fuels, bituminous coal, No. 6 fuel oil, and natural gas. The respective percentages of hydrogen in these three fuels are 5.0, 11.5, and 27.5 per cent. The chart shows overall combustion and furnace efficiencies; the stack loss is the difference between 100 per cent and the indicated efficiency.

From the foregoing discussion the need for giving serious consideration to arranging the combustion equipment so that correct mixtures of fuel and air can be automatically maintained under varying loads and varying draft conditions, and so that continuous readings or results may be taken to check the plant operation, is evident. The days of depending on the fireman to judge efficiency by looking at the fire have passed. Even if he could judge combustion efficiency by visual inspection, which is improbable, he would have to be changing valves and dampers constantly to compensate for varying conditions.

And Now—The Air Conditioned Billboard!

AIR conditioning continues to find new applications—witness the unique Wilson & Co. billboard on Michigan Ave., Chicago, dedicated last month. Into it are built air conditioned, plate glass kennels in which are exhibited pedigreed dogs.

Air conditioning is necessary because the board faces south and has a large glass area. The system, designed to maintain 80 F and 50 per cent R. H. with outside conditions of 95 F D.B. and 75 F W.B., consists of a 5 hp condensing unit, two conditioner units so installed as to permit a mixture of 50 per cent outside and 50 per cent recirculated air, an atmospheric cooling tower with a 2 hp pump circulating 25 gpm of condenser cooling water, hair mat air filters, and automatic temperature control. Total amount of air circulated is 2000 cfm.

Walls and ceiling of the kennels are insulated with animal hair insulation. The floor is slotted, and as the interior is under pressure, the vents under the slotted floor eject air at the rate of 1000 cfm, maintaining healthful atmospheric conditions in the kennels. The conditioning equipment is in operation 24 hours a day, the compressor and circulating pump operating under thermostatic control. The air conditioning installation was made by R. Cooper, Jr., Inc.

WILSON & CO.
WILSON & CO.
"O.K. WILSON & CO. YOUR QUALITY CONTROL METHODS ENTITLE YOU TO MY CERTIFICATE"
Ideal DOG FOOD
An ideal meal FOR YOUR DOG
Ideal DOG FOOD

Pressure Losses in Rectangular Elbows

By R. D. Madison* *and* J. R. Parker†

The authors present data on pressure losses in rectangular elbows as affected by (*a*) the curve ratios of the elbows, (*b*) the aspect ratios of the elbows, (*c*) angle of the bend, (*d*) the elbow size, (*e*) velocity of flow, (*f*) splitters in the elbows, and (*g*) compound elbows. In general, data are given (1) for the elbow situated in a duct system with ducts preceding and following the elbow, and (2) the elbow at the end of a duct and discharging freely into the air

[PART 1]

PROBABLY the earliest comprehensive tests for the determination of the pressure loss in sheet metal elbows were made on 12 in. square and 12 in. diameter elbows of varying radii by F. L. Busey.[1] More recently Loring Wirt[2] published results of tests which included the effect of rectangular shaped elbows. Both experimenters seem to have overlooked the fact that surface friction varies with the size and shape of the elbow and in turn has a direct bearing upon the total pressure loss. With this thought in mind the authors present the results of tests on rectangular elbows in which the effects of the size and shape of the elbow and the surface friction are considered in the determination of the pressure losses.

Definition of Terms

For the purpose of this paper, it will be convenient to refer to all elbows of the same size as having the same cross-sectional area. The velocity V through the elbow or duct will be the average velocity, *i.e.*, the quantity of flow divided by cross-sectional area.

The radius ratio is designated as RR and defined as the radius of the center line of the elbow divided by the depth in the plane of the bend, *i.e.*, divided by the difference between the outside and inside radii. In order to derive the most value from the data plotted, a new term will be used in addition to radius ratio. This term will be referred to as the curve ratio, designated as CR and defined as the inside radius of curvature divided by the outside radius of curvature. It is to be noted that in the use of the curve ratio instead of the radius ratio as a coordinate in plotting curves, (1) zero radius ratio is eliminated, (2) the curves are separated more effectively, and (3) curves can be drawn to include the complete range of radius ratio values from 0.5 to infinity. Numerically, the following relations exist between the curve ratio and the radius ratio in this region

$$CR = \frac{RR - 0.5}{RR + 0.5} \text{ and } RR = 0.5\,\frac{1 + CR}{1 - CR}$$

It will be observed that zero radius ratio referred to by Busey[1] does not belong to the same family of curves as those plotted using the curve ratio, the limiting condition being 0.5 radius ratio (zero curve ratio) where the inside corner of the elbow is a sharp edge and the outside radius is formed with this edge as a center.

The aspect ratio AR is the width of elbow along the axis of bend divided by the depth in the plane of the bend. High aspect ratios will therefore characterize "easy" bends and low aspect ratios will characterize "hard" bends.

The term "duct friction" will be referred to as the normal friction of straight ducts or that portion of an elbow loss which corresponds to duct friction. Similar to the losses in a duct, the losses in an elbow will consist of (1)

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[1] "Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes and Ducts," by F. L. Busey, *Transactions, American Society of Heating and Ventilating Engineers*, Vol. 19, 1913, p. 366.

[2] "New Data for the Design of Elbows in Duct Systems," by L. Wirt, *General Electric Review*, Vol. 30, June, 1927, p. 286.

Contributed by the Aeronautic Division and presented at the Annual Meeting of *The American Society of Mechanical Engineers* held in New York, N. Y., December, 1935. Published in *A. S. M. E. Transactions* for April, 1936.

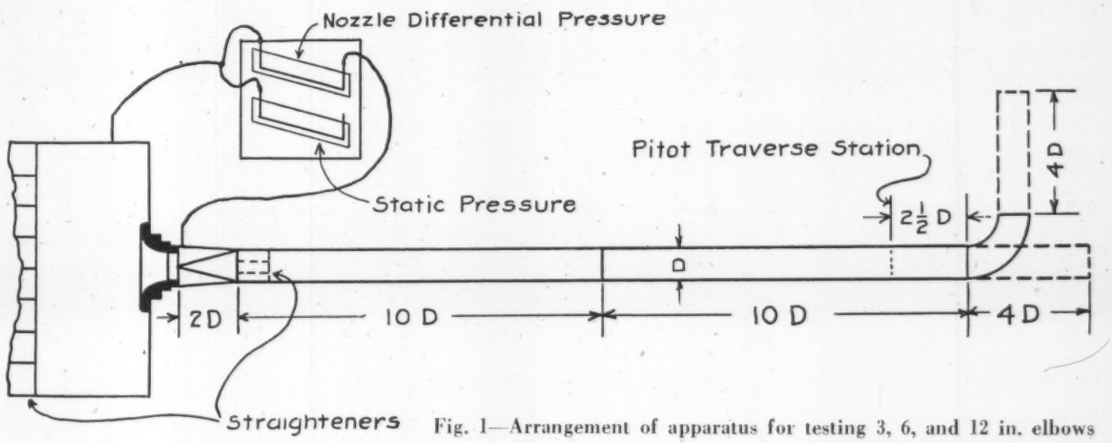

Fig. 1—Arrangement of apparatus for testing 3, 6, and 12 in. elbows

eddy loss due to the internal friction of the fluid itself and (2) film friction where the fluid is in close proximity to the surrounding walls. The eddy loss will vary substantially as the second power of the velocity while the film friction will vary as the first power of the velocity. Duct friction will, therefore, have both components and, in proportion to each, the combined loss will vary as a power, usually between 1.75 and 1.85. "Shock loss," in this connection, will be referred to as that loss over and above normal duct friction. All losses are measured as a percentage of the pressure corresponding to the mean velocity in the elbow.

Apparatus and Methods

Preliminary tests were made on elbows with cross-sectional areas of 9, 36, and 144 sq in. by J. R. Parker in his own laboratory and were checked by the coauthor in the laboratory of the Buffalo Forge Co. and extended to include other sizes of elbows. It was realized that a prohibitive amount of time would be involved in making the tests unless simple procedures were employed. This is especially true in conducting tests with large-sized elbows in which a 35 point traverse was made. In the early tests on small elbows the straight duct ahead of the elbow was traversed and a calibration arrived at for a pitot tube placed in the center of the duct 2½ diameters ahead of the elbow. This was later checked by the use of a calibrated nozzle as shown in Fig. 1. For the large-sized elbows, a special pitot grid was constructed. This consisted of a framework carrying 35 symmetrically placed total pressure tips which were interconnected. Upon the same framework there were 10 static pressure tips also interconnected. The velocity pressure thus recorded by the difference was substantially the average pressure instead of the pressure corresponding to the average velocity. This difference, however, was from 1 to 3 per cent of the velocity pressure and was substantiated by and appeared in the calibration with the standard traverse. Straighteners were used in all cases and sensitive, inclined, oil filled manometers readable to 0.001 in. water pressure were employed.

When air is discharged directly from an elbow, the velocity over the outer section is higher than normal and contains a double spiral component of motion. When the elbow is followed by a duct of 3 to 4 diameters in length, there is a partial regain in static pressure as the velocity is again lowered to normal and the spiral movement subsides. The net effect, therefore, is for the elbow loss to be lower when the elbow is followed by a straight length of duct. The pressure loss was measured by testing the standard duct with a short length (equal to four diameters of duct) attached at the end. Then a test was made at the same air velocity in the duct by interposing the elbow between the standard duct and the short length, as indicated in Fig. 1. The difference in the two static pressure readings taken at the same point ahead of the elbow, when divided by the velocity pressure, gives the elbow loss in percentage of the velocity pressure. This procedure gives the elbow loss "followed by a duct" but does not include the duct loss of the short length itself. Where the elbow loss "only" was desired, that is, the elbow discharging into free air, the short length of duct was omitted from both tests. For square ducts, an aspect ratio of unity, the elbow was rotated to each of the four possible positions and the readings averaged. With other aspect ratios, both of the possible positions were tested and the readings averaged. Theoretically, there should be no difference in the readings when the elbows are tested in the different positions; actually, there were small differences especially when elbows of small radius ratios were not followed by a straight length of duct and also when larger-size elbows were attached to a main duct leading from a fan with the same outlet dimensions.

Effect of Radius Ratio

The effect of curve ratios may be seen by referring to Fig. 3. Curve *A* is plotted from data obtained by Busey[1] on 12 in. × 12 in. elbows. Curve *B* is plotted from data obtained by O. E. Parker[3] on 12 in. × 12 in. elbows. Curve *C* is plotted from data obtained by J. R. Parker, during tests of identical elbows such as shown in Fig. 2. Curve *D* is plotted from data obtained by Wirt[2] on 3 in. elbows. Points *E* and *F* are from data obtained by Madi-

[*Continued on page* 369]

[3] "An Investigation of Pressure Losses in Air Duct Elbows," by O. E. Parker, presented in 1934 as a thesis for the degree of B. S. in mechanical engineering, Northeastern University.

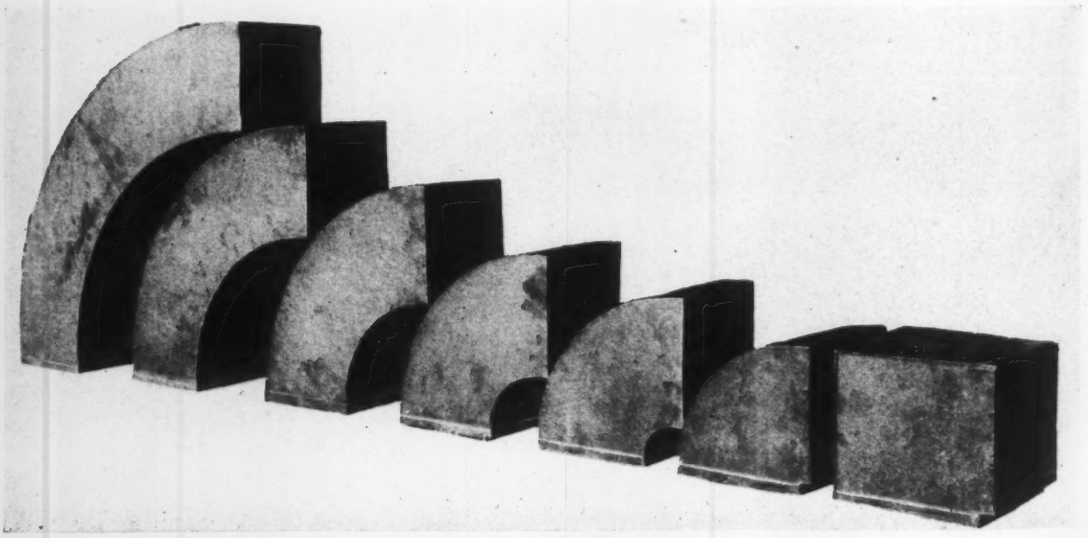

Fig. 2—Elbows of various radius ratios

Holding Properties of Gaskets Studied

[A translation* by F. E. Wertheim, Member, A. S. M. E. and W. S. E., of a report of a German investigation, with a discussion by the translator]

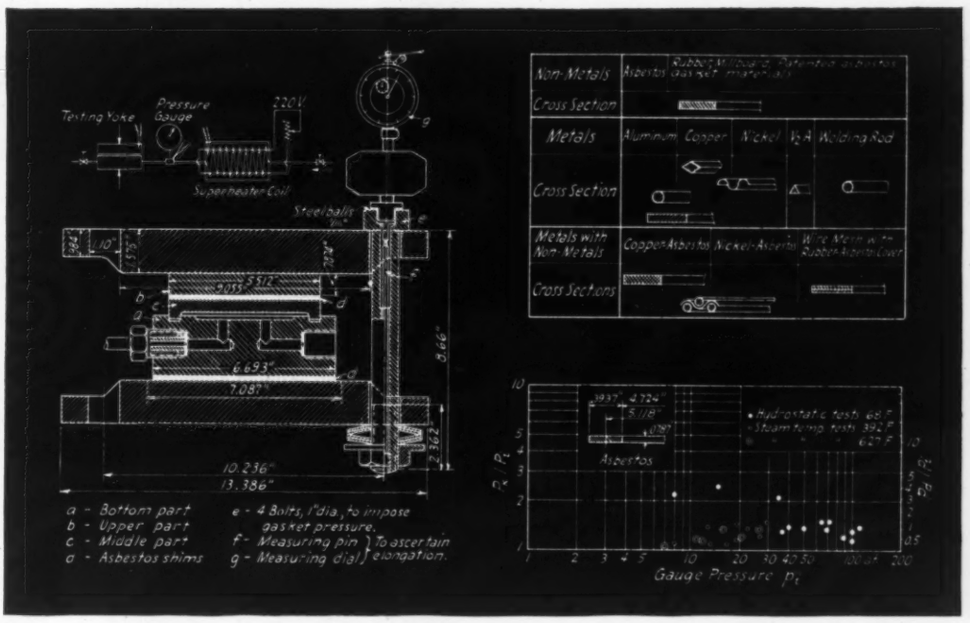

Fig. 1 (*left, above*) and Fig. 2 (*left, below*)—Arrangement of apparatus for experimental investigation of holding properties of gaskets subjected to steam heat. Fig. 1 is a schematic diagram and Fig. 2 a detail of test apparatus. Fig. 3—(*right, above*)—Investigation of gasket materials and shapes. Fig. 4—(*right, below*)—Experiments with non-metallic gasket materials

IN order to obtain basic information concerning the gasket pressures necessary to assure tightness with various gasket shapes and materials at designated pressures, the holding properties of these materials were investigated at room temperature by means of hydrostatic tests; later, the gaskets were tested under steam pressure at temperatures up to 350 C (662 F).[1]

The arrangement utilized for the hot tests is illustrated in Figs. 1 and 2. First, steam at 35 atmospheres gauge flowed through a control valve; it could be raised to a temperature of 600 C (1112 F) in an electrically heated superheater. It was then conducted to a measuring station where pressure and temperature were determined, next it traversed the test apparatus and finally escaped to the atmosphere through a second control valve. The test apparatus, Fig. 2, consisted of a steam traversed bottom piece *a* with a circular groove 10 mm (0.394 in.) wide for the reception of the gaskets, upon which the top piece *b* was pressed by the calibrating bolts *e*, by means of which the load could be determined from bolt stretch, taking into account the elongation due to temperature. In the gasket tests the usual procedure was first to place these under bolt load in the yoke apparatus, heated by steam flowing through it; then a definite steam pressure was imposed; after which the bolt load was successively decreased until leakage became apparent.

In the manner described, an entire series of gasket materials and shapes was tested, which are assembled in Fig. 3.

For computation of flanged connections the presentation of experimental results would appear advantageous, since from them we may derive the relation between the residual force P_k carried by the assembly and adequate to prevent leakage and the force P_i, produced by unit fluid pressure, p_i. In addition, the relation between the unit gasket pressure, p_d and the unit fluid pressure, p_i, (contact-pressure ratio) may be worth noting. In Figs. 4 to 8 these relations are given for several characteristic gasket types. Through use of double logarithmic paper the experimental values representative of lower pressures are thereby brought into stronger relief.

The non-metallic gasket materials such as asbestos or those materials consisting of asbestos with special bonding mediums show low values for the relation P_k/P_i, (see Fig. 4) at room temperature as well as at 200 to 350 C

*Translated from summary by E. Seibel in *Zeitschrift* des Vereines Deutscher Ingenieure May 5, 1935, pp. 556-7.

[1] E. Seibel, W. G. Hering and A. Raible, Forschg. Ing.-Wes., Bd. 5 (1934) S. 298.

(392 to 662 F). Throughout the above experiments this relation is less than two. We should take into account, however, that the width of the gaskets with 120 mm (4.73 in.) inside diameter amounted only to 10 mm (0.394 in.). For this group of gasket materials the gasket pressure necessary for sealing, for both the room temperature and hot tests, corresponds approximately to the internal fluid pressure ($p_d/p_i = 1$). After the experiments at 350 C (662 F) some of the gaskets were found to be partially destroyed due to burning out of constituents within the bonding medium.

Discussion

The subject of optimum gasket pressure has long been a fertile field for imagination and argument. Designers faced with problems involving high temperatures and pressures, often associated with corrosion, have found that machine design textbooks contain meagre information. Hence, many of these men, lacking the necessary experience, have had to rely on intuition, rather than on the fruits of research. The work of Siebel, Hering and Raible comes as a timely and very useful contribution.

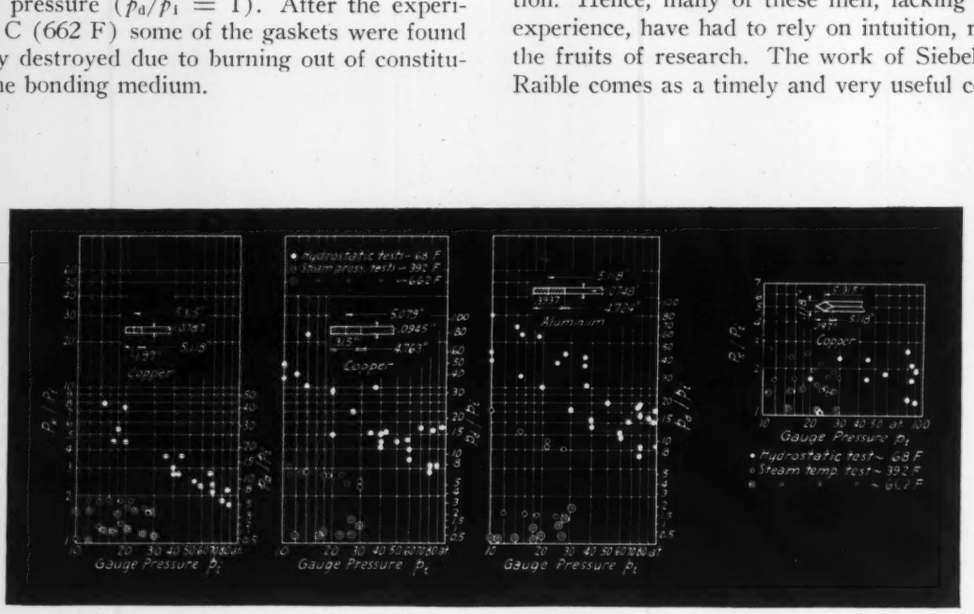

Fig. 5 Fig. 6 Fig. 7 Fig. 8

Figs. 5-8—Experiments with copper and aluminum gaskets of rectangular cross-section, and copper gaskets with diamond shaped cross-section, at varying temperatures, depending upon unit fluid pressure, p_i. The points indicated are those experimental values for steam pressure and bolt load with which leakage ensued. p_i is unit fluid pressure, p_d unit gasket pressure, p_k residual force = resultant or total gasket load, P_i total fluid pressure load due to p_i

With the metallic gasket materials it was found that at room temperature, the necessary gasket pressure is a multiple of the internal fluid pressure; correspondingly the values P_k/P_i, particularly with low pressures, lie extraordinarily high. According to Figs. 5 and 6, however, the gasket pressure for copper at steam temperatures of 200 C (392 F) sinks to a fraction of the value necessary at room temperature. This may be explained because, even at 200 C, the resistance of copper to deformation diminishes noticeably, so that at higher temperatures the copper gaskets gradually adapt themselves to the flange faces under decreased loading.

Profile metal gaskets possess an advantage over flat gaskets, as shown by a comparison of Figs. 5 and 6 with Fig. 8, because here the ratio P_k/P_i which is requisite for tightness is lessened, particularly with lower internal fluid pressure. A result similar to that of profile gaskets may be attained by corrugating metal gaskets.

From these experiments we may conclude that for every pressure and every temperature, gaskets may be found with which satisfactory fluid tightness under working conditions is assured when the total resultant force imposed upon the gasket amounts to at least double that attained by the force due to internal fluid pressure.

The following comment is based upon study of the complete investigation referred to in Footnote 1, rather than upon the short abstract translated above.

The apparatus employed in making the hydrostatic tests, while similar to that shown in Fig. 2, was squeezed in a press instead of being compressed by the action of bolts. The reason for this difference in method would be interesting, but is not stated. Credit for the idea of determining gasket pressures in this manner probably belongs to the German engineer, Hermann Haeder[2]. Following Haeder, Sandstrom[3] has suggested a method of ascertaining gasket pressures which is practically identical with Siebel's.

Fig. 4 is characteristic of results obtained from the non-metallic gasket materials tested, which included rubber, fiber, leather, asbestos, and several special asbestos-rubber compounds.

In contrast with the numerous non-metallic materials, very few metal gaskets were tested, the plain ring gaskets of sheet metal including only copper, aluminum and lead; not one of which should be specified for use with super-

[2]See "Construieren & Rechnen," 43f, 6th ed., 1914.
[3]*Chemical and Metallurgical Engineering*, December, 1932, p. 671.

heated steam. It is a matter of regret that this research does not include tests of soft annealed sheet steel and monel gaskets, which *are* specified for high temperature and pressure steam and are, besides, in much more general use than either copper or aluminum. The report also informs us that the aluminum gasket employed for Fig. 7 was cut from a cold rolled sheet, one of the characteristics of this material being that recrystallization sets in at 300 C (572 F). We may state in this connection that aluminum gaskets are quite satisfactory for steam at temperatures up to 260 C (500 F), and that for such purposes "2SO" aluminum should be specified. (The "2S" is commercially pure metal, the "O" means annealed dead soft.) The strength of copper is likewise seriously affected above a temperature of 260 C, thereby limiting its use for gasket purposes.

The subject of optimum gasket pressures has long been a fertile field for imagination and argument. Consequently, it is felt that this summary of the results of a German investigation made to obtain basic information concerning the gasket pressures necessary to assure tightness with various gasket materials and shapes will interest piping engineers

In comparison with the foregoing, soft annealed sheet steel is cheap and it is entirely satisfactory even at the highest temperatures. Monel, too, is much used in special cases, and it, also, is suitable for high temperatures.

Profile solid serrated metal gaskets are discussed, and since this is a standard type the omission of any test results for them makes the report appear incomplete.

Although Siebel investigated different widths of each material (the ratios $\frac{\text{Outside Diameter}}{\text{Inside Diameter}} = \frac{D_a}{D_i}$ ranged from 1.17 to 1.75) definite recommendations for proportioning gaskets are scarcely to be expected, due to the comparatively small difference in width. It is of interest that Sandstrom[4] states "Tests *seem to indicate* that the resistance to compression (of the gaskets) varies directly with the square of the width and (nearly) inversely with the thickness."

Gasket thickness is an important matter, governed in the piping code by the stipulation: "Shall be as thin as the finish of surfaces will permit. . . ." It would seem, therefore, that tests of some thinner gaskets, say on the order of 26 ga (0.019 in.) might yield useful data. The question also arises as to what pressures might have been required had the gaskets been omitted and the metal-to-metal joint (formed by the parts of the apparatus) tested.

We hope, as E. O. Waters, D. B. Wesstrom and Frank S. G. Williams suggested in the December, 1934, *Mechanical Engineering*, that those who possess the necessary experience will offer data relating to "contact-pressure ratios," and that research will further clarify the problem.

[4] *Chemical and Metallurgical Engineering*, March, 1934, page 133.

[*Continued from page* 366]

son on 12⅜ in. and 33 in. elbows, respectively, and point *G* from data obtained by Brown[4] on a 12 in. elbow. Reverse curve *H* in Fig. 3 shows the theoretical duct loss of a 12 in. × 12 in. elbow based upon a factor of one velocity head loss in 55 diameters of straight duct, one diameter being equivalent to the side of a square duct. Note that in such a graph as Fig 3, the extreme right abscissa which is unity represents an elbow of infinite radius and indicates that the loss in a 90 deg elbow would be the same as in an infinite length of straight duct. The complete curve of elbow loss will, therefore, follow down along curve *C* as the radius ratio increases and then cross the space shown approximately by the dashed line and blend into the friction curve *H*. A value greater than 2 for a radius ratio (0.6 curve ratio) is therefore of questionable merit.

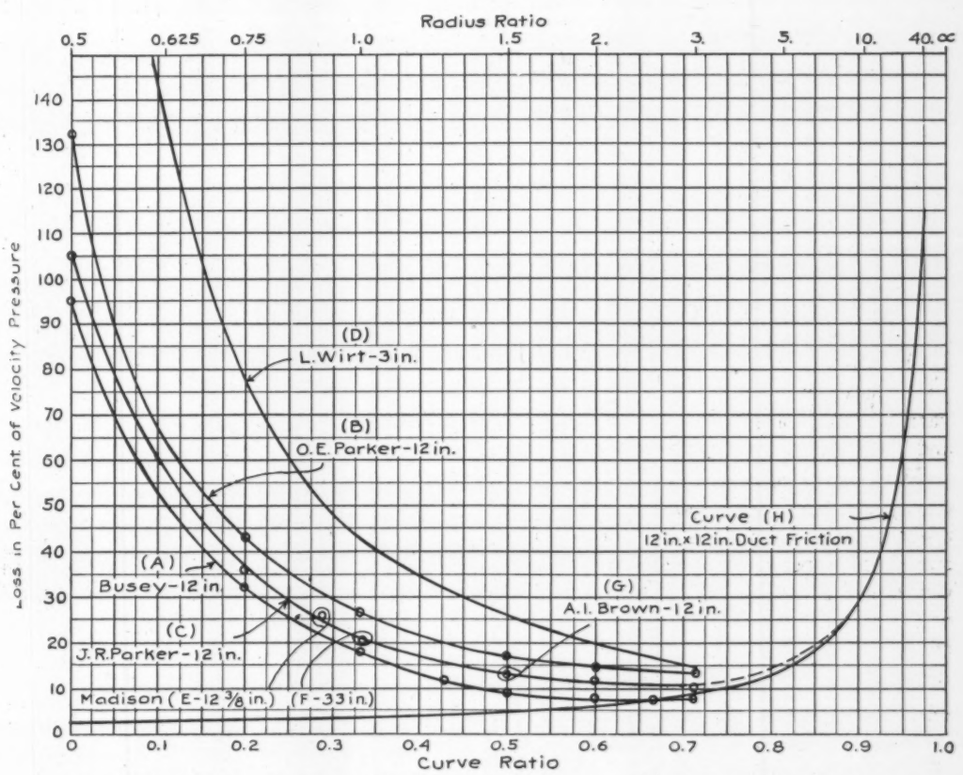

Fig. 3—Pressure loss in 90 deg elbows followed by ducts, using curve ratios as abscissas

The similarity of curves *A*, *B*, and *C* in Fig. 3 probably indicates the true trend of the pressure loss.

[*To be continued*]

[4] "Friction of Air in Elbows," by A. I. Brown, *Power Plant Engineering*, Vol. 36, August 15, 1932, p. 630.

High Speed Synchronizing Motor Control *for Air Conditioning Compressors* Insures Minimum Lighting Disturbance

By R. C. Allen*

To the many services performed by the electronic tube is now added its function in air conditioning for synchronizing motors driving compressors at definite pre-selected crank angular displacements to insure minimum power and lighting disturbance....The author describes the necessity of synchronizing compressor motors, explains the operation and application of the equipment

SYNCHRONOUS motor driven compressors for air conditioning are supplied, as standard, with sufficient flywheel effect to limit the current variation resulting from the varying torque during a revolution to 66 per cent of the motor full load current rating, which is usually ample in an ordinary power distribution system to prevent any disturbance on the lighting system.

A current variation produces a change in the transformer and line drop and, therefore, a change of voltage at the lamp. The objection to a change of light intensity depends upon the actual voltage variation and its frequency of occurrence as well as the individual and class of work being done. Fig. 1, compiled from many tests conducted by power companies and others, indicates the maximum voltage variation at different frequencies for an ordinary lamp which would be perceptible to a sensitive eye.

Where the compressor load is a large proportion of the line capacity and the lighting is fed from the same transformers—as is frequently the case—a pulsating load will cause the greatest disturbance; the greater the proportion of pulsating load, the greater will be the disturbance. With two or more compressors, staggering the crank positions between the units helps to iron out the current variation and causes less disturbance than if the cranks and current peaks came in phase.

The cranks of the two cylinders of horizontal double acting carbon dioxide compressors are usually 180 deg apart in order to reduce the size of the flywheel for the standard 66 per cent current variation. This, however, causes gas pulsations in the piping every half revolution, resulting in movement of the piping unless it is properly anchored. In fact a large South American nitrate plant reported that occasionally the synchronizing of these gas pulsations caused the whole building super-structure to pulsate, and it was necessary to shut down and restart one or more compressors to prevent serious trouble.

To stagger compressor cranks satisfactorily under the normal method of starting is almost impossible and, until the advent of electronic tubes, there was no equipment that would function accurately enough in the small space of time available. Synchronous motors are designed with induction motor torques that will bring the load up to about 95 per cent speed from which point the motor is synchronized by exciting the field winding. With a 32 pole motor the synchronous speed is 225 rpm. At 5 per cent slip the induction motor speed is 213.8 rpm. The rotor is therefore slipping one set of poles (360 electrical degrees) each 1/8 second and to synchronize successfully, the poles should not be more than 180 deg (electrical) out of phase position, Fig. 2.

Two cylinder horizontal double acting compressors with cranks at 90 deg should have cranks displaced by some 45 deg angle (135 deg or 225 deg) for two compressors. Keyways can be located so that this is possible, but failure of the control to function accurately would cause the motor to synchronize too soon or too late by at least a pair of poles or 22.5

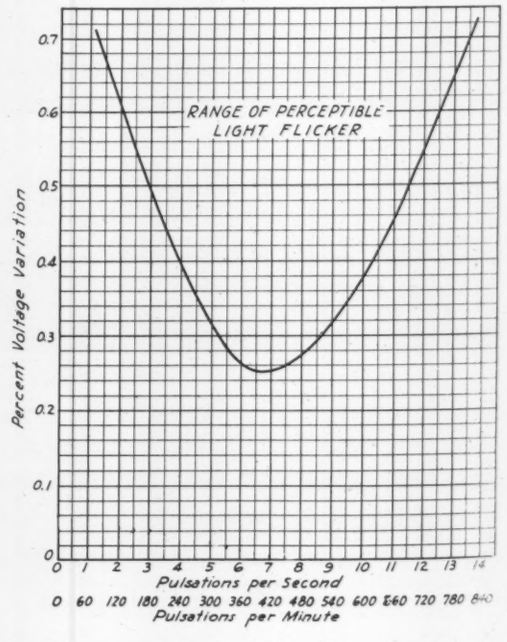

Fig. 1—Curve compiled from various tests showing voltage change at various frequencies or pulsations—as from a reciprocating compressor—at which light flicker is just perceptible

*Engineering Div., Westinghouse Electric & Mfg. Co., Philadelphia, Pa.

deg (mechanical) from the desired position for motors of the above speeds.

The time required for the usual relays and contactors to function from any contact device on the shaft would be too long and too uncertain to be satisfactory. This difficulty has been overcome by use of inductive generators on the motor shafts and grid glow type tubes and high speed relays in the field closing circuit. The equipment can be used in conjunction with standard synchronous motor control, the control functioning in the ordinary manner up to the point of applying the excitation voltage; instead of closing the field contactor at that point, the high speed synchronizing equipment is energized. This in turn applies the excitation to the motor the first time the rotors of the two machines are in the correct angular position relative to each other.

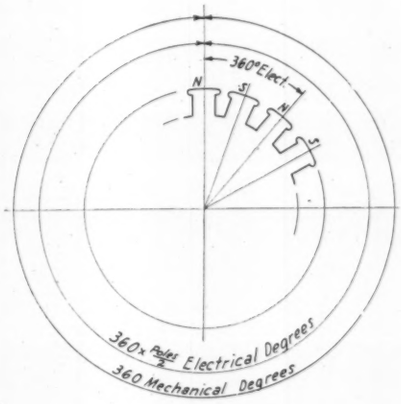

Fig. 2—Relationship between electrical and mechanical degrees

For 225 rpm and 60 cycles, number of poles = 32, pair poles = 32/2 = 16. 1 pair poles = 360 electrical degrees. 1 revolution or circumference = 360 × 32/2 = 5760 electrical degrees = 360 mechanical degrees. 1 pair poles = 360° ÷ 32/2 = 22.5 mechanical degrees.

Fig. 3—High speed synchronizing control used in connection with motor starters for reciprocating compressors for air conditioning Kansas City Auditorium to obtain a definite angular relationship between compressor cranks and thus reduce the effect of the pulsating load current on the building lighting system

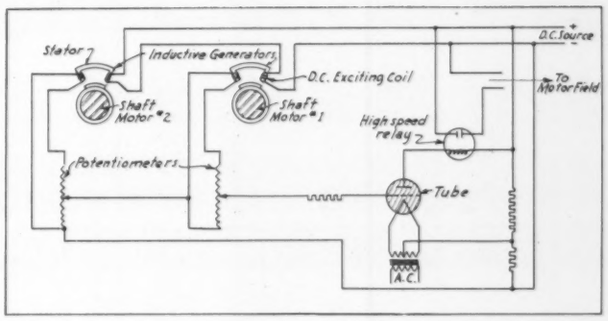

Fig. 4—Schematic diagram of high speed control for applying synchronous motor field at a definite rotative position with respect to a motor running at synchronous speed

The Installation at Kansas City Auditorium

To insure minimum power and lighting disturbance, such equipment was supplied with the electrical control for the four 400 hp, 225 rpm, 199 volt, 3 phase, 60 cycle, 100 per cent power factor synchronous motors driving four cylinder vertical single acting compressors for air conditioning the Kansas City Auditorium. These are arranged in pairs with the control so designed that either motor can be started first and the second synchronized at the desired crank position.

An inductive generator was installed on each shaft. The tube control (inset, Fig. 3) consists essentially of a potentiometer in each inductive generator circuit and a tube of the grid glow type together with high speed relays for applying the excitation to the motors. Fig. 4 is a schematic diagram of this control. These induction generators have no rings, brushes, or wearing parts. The rotor is a small steel sector carried on an insulating ring attached to the shaft. The stator is an electromagnet attached to the brush rigging support. The generator is excited from a direct current source and the generator winding is connected in the grid of the tube.

As the rotor passes under the poles of the electromagnet, a voltage is induced which is impressed on the grid of the tube with such a polarity as to reduce its negative

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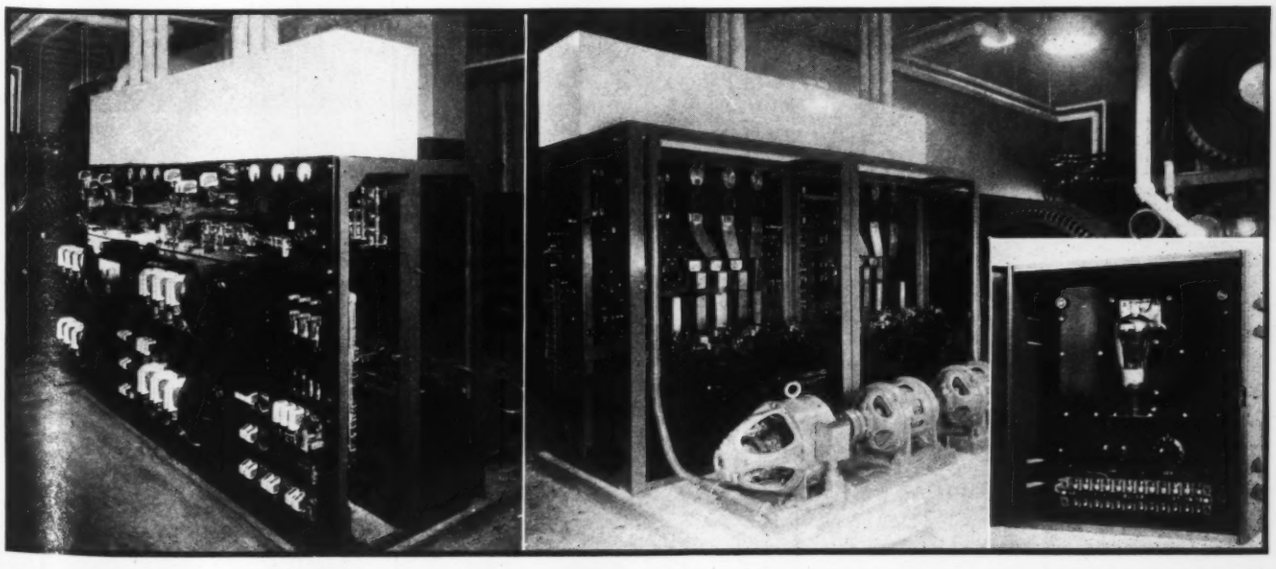


Friction Effect of Slip Joints, Rollers and Guides on Long Steam Transmission Pipe Lines

Results of an Investigation on an 8,000 Ft, 12 In. Main

LINEAR EXPANSION OF SEAMLESS O.H. STEEL PIPE
DATA SHEET

OPERATING TO NON-OPERATING CHANGE
(Hot to Cold)

Straight Run of Pipe Feet	Avg. Non-Operating Temp. Deg. F	Avg. Operating Temperature Deg. F.	Temperature Difference Deg. F.	Total Elongation Feet	Actual Coefficient of Expansion
1065.5	67.5	513.9	446.4	3.282	.00000690
942.7	65.8	495.1	429.3	2.802	.00000692
1523.0	69.0	460.4	391.4	4.083	.00000685
1020.8	68.2	416.0	347.8	2.428	.00000684
1725.5	67.8	395.5	327.7	3.863	.00000683

NON-OPERATING TO OPERATING CHANGE
(Cold to Hot)

1065.5	67.5	425.5	358.0	2.634	.00000690
942.7	65.8	414.0	348.2	2.259	.00000688
1523.0	69.0	409.0	340.0	3.565	.00000688
1020.8	68.2	403.0	334.8	2.344	.00000686
1725.5	67.8	393.0	325.2	3.845	.00000685

Actual average coefficient of expansion ..... .00000686
Coefficient from Smithsonian Physical Tables ..... .00000688

RESULTS of an investigation conducted to determine the effect of slip joints, rollers, and guides on the linear expansion of a long straight run of pipe due to an increase in temperature, were discussed by Claude R. Erickson* in the research committee's report presented at the *National District Heating Association's* convention last month.

It was determined that in a steam transmission or distribution main, the resistance offered by well designed and maintained slip joints, rollers and guides has no effect on the elongation of the line due to an increase in temperature. The elongation of the pipe line may therefore be accurately computed as pure linear expansion. If the actual elongation of the pipe as measured at the slip joint is less than the computed elongation it may be attributed to one of the following reasons: (1) The use of an incorrect coefficient for the pipe material under consideration. (2) Movement of the anchor supports. (3) Transverse deformation of the pipe.

Metals constituting pipe have the property possessed by all materials of expanding with an increase in temperature. In the construction of a long steam transmission or distribution main, allowance for this expansion may be provided for either by suitable joints or bends. Although the expansion bend is free of the pressure and temperature limitations of joints it has the disadvantages of added friction loss, the greater expense of fabrication, and the increased space requirements. Originally these bends were designed by empirical rules but in the last few years a stress strain study of pipe bends on the basis of structural theory has been made. The slip joint, at the present time, is the most generally used means of providing for expansion in long mains.

*Engineer, Board of Water Electric Light Commissioners, Lansing, Mich.

References may be found to the effect that the linear expansion of a pipe calculated from coefficients of expansion arrived at from laboratory determinations on short lengths of test pieces has provided for more expansion, as measured at the joint, than actually occurred in the field.

To investigate this possibility a 12 in. steam transmission main approximately 8,000 ft in length and designed for 400 lb pressure recently installed was provided with accurately placed measuring lugs at each slip joint. After the entire line had been installed and the trench back-filled a sufficient length of time to permit the pipe to reach a normal non-operating temperature, these lugs were set with a templet and welded to the pipe a distance of 2 ft from the face of each joint. Because this line was intended for seasonal operation these lugs provided a means for accurately determining movement of any of the anchor supports. Measurements taken after the operating season would determine whether or not the main was in its original non-operating position. Accurate measurements of the elongation as measured at the slip joints and corresponding line temperatures were taken over a temperature range of 450 F.

If the elongation of the pipe were unrestrained and occurred in a line parallel to the center line of the pipe, then the increase in length due to the increase in temperature could be found by the following formula for pure linear expansion:

$$l = KLt$$

where

l = Change in length of pipe L ft long, ft
t = Temperature change, F
K = A constant

Although there is considerable variation in the coefficient of expansion as reported from different sources the data of Holburn and Day, Smithsonian Physical Tables, 7th edition, are recommended by reliable authorities and were used in the comparisons. The coefficients for metals generally used in pipe are:

Charcoal iron 0.00000686
Bessemer steel 0.00000699
Seamless open hearth steel 0.00000688

The coefficient calculated from the actual elongation of the seamless open hearth steel main described above was 0.00000686.

Solving the Four Types of Process Air Conditioning Problems

By William Goodman*

All air conditioning problems fall into one of four main divisions, depending upon whether the sensible and latent heat are being subtracted from, or added to, the space being conditioned. Problems typical of each division are discussed here and solutions by the psychrometric chart explained

[Part 2]

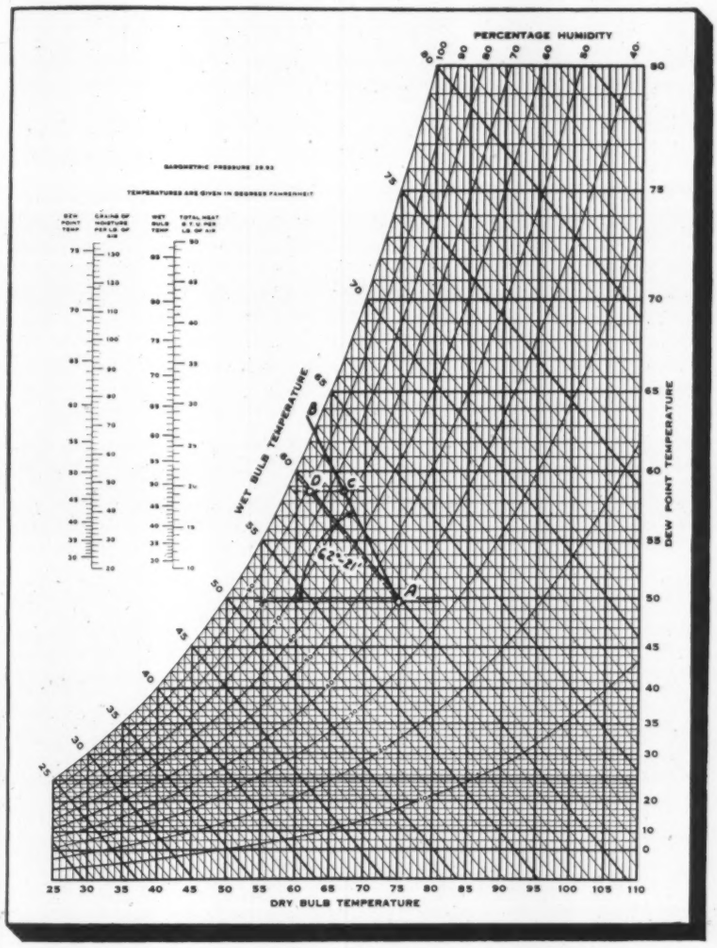


Fig. 4—Solution of Example 3, cooling and humidifying

IN Part 1[1], the four main divisions into which air conditioning problems fall, depending upon whether the sensible and latent heat are being subtracted from or added to the space being conditioned, were described, and Examples 1 and 2 were presented. Part 2 concludes the discussion.

Cooling and Humidifying

Under some conditions the numerical value of the latent heat loss may be greater than the sensible heat gain. When this occurs, the value of the sensible heat ratio will be negative instead of positive, and the heat ratio line will lie above the line of room wet bulb temperature. In Example 2,[1] where the sensible heat gain was greater than the latent heat loss, the ratio line was below the line of room wet bulb temperature.

EXAMPLE 3—A room which is to be maintained at a dry bulb temperature of 75 F has a sensible heat gain of 49,000 Btu per hr and requires the addition of 71 lb per hr of moisture in order to maintain its relative humidity at 40 per cent. If the air supply is to have a dry bulb temperature of 67 F, find its required wet bulb temperature.

Solution—

Latent heat loss = $71 \times 1050 = 74{,}600$ Btu per hr

$$\text{Sensible heat ratio} = \frac{49{,}000}{49{,}000 - 74{,}600} = \frac{49{,}000}{-25{,}600} = -1.92$$

Referring to Table 3,

Heat ratio angle = 62° — 21′

The point *A* of Fig. 4 represents the required room condition of 75 F and 40 per cent relative humidity. The line *A-B* is the heat ratio line. The required wet bulb temperature of the air supply, as read at the point *C* on the line *A-B* is 61.8 F.

Notice that in Fig. 3, for Example 2,[1] the heat ratio line is below the line of room wet bulb temperature while in Fig. 4, for Example 3, the ratio line is above the line of room wet bulb temperature. Interpreted practically this means that, in the latter case, a spray of warm water

*The Trane Co., LaCrosse, Wis. Copyright, 1936, by William Goodman.

[1] HEATING, PIPING AND AIR CONDITIONING, June, 1936, pp. 323-325.

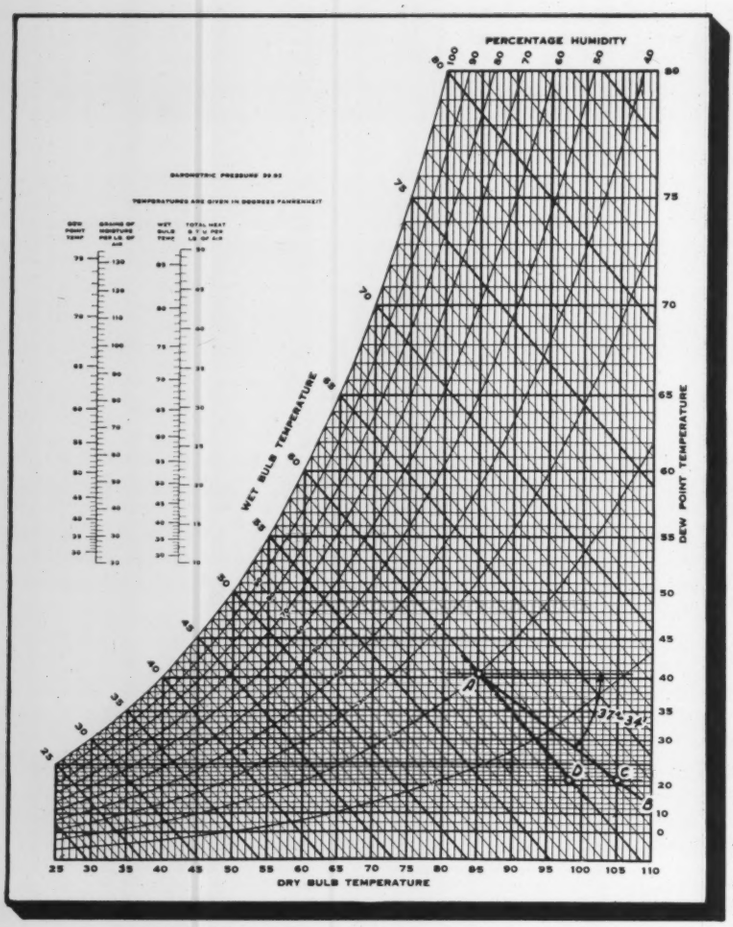


Fig. 5—Solution of Example 4, heating and dehumidifying

would be required to condition the air supply, while a spray of chilled water would be required for the conditions of Example 2. The air supply could also be humidified by a water spray having a constant temperature equal to the room wet bulb temperature. The air would be humidified along a line of constant wet bulb temperature from point *A* to point *D*, Fig. 4. After leaving the spray, the air could then be heated by a coil along a line of constant dew point temperature such as *D-C* to bring it to the required dry bulb temperature for the conditions of Example 3. For the conditions of Example 2, it would be necessary to cool the air supply after leaving the sprays. The air would be cooled by a coil along a line of constant dew point temperature, such as *D-C* of Fig. 3, to bring it down to the required dry bulb temperature for the conditions of Example 2.

Heating and Dehumidifying

Some drying processes are carried on in an atmosphere of high temperature and low humidity; as a result, there is a loss of sensible heat from the conditioned space and at the same time a continuous gain of moisture from the objects being dried. Therefore the air supply must surrender sensible heat and simultaneously absorb moisture. To accomplish this, the initial condition of the air supply must lie in the fourth quadrant.

EXAMPLE 4—A dry bulb temperature of 85 F and a relative humidity of 20 per cent are to be maintained in a conditioned space. The sensible heat loss through the walls of the drying room is 135,000 Btu per hr. Eighty lb per hr of water are to be evaporated from the objects being dried. The latent heat of evaporation is provided directly by steam coils on which the wet objects rest during the drying process. Find the required wet bulb temperature of the air supply if its dry bulb temperature is to be 105 F.

Solution—

$$\text{Latent heat gain} = 80 \times 1050 = 84{,}000 \text{ Btu per hr}$$

$$\text{Sensible heat ratio} = \frac{-135{,}000}{-135{,}000 + 84{,}000} = \frac{-135{,}000}{-51{,}000} = 2.65$$

Referring to Table 3,

$$\text{Heat ratio angle} = 37^\circ - 34'$$

The point *A* of Fig. 5 represents the required room condition. The line *A-B* is the heat ratio line. The required wet bulb temperature of the air supply as read at the point *C* is 62.2 F.

In this case, when the sensible heat loss is greater than the latent heat gain, the heat ratio line will fall above the line of room wet bulb temperature. When the sensible heat loss is less than the latent heat gain, the heat ratio line will fall below the line of room wet bulb temperature.

An air supply whose condition is represented by a point in the fourth quadrant, may be obtained in one of two ways; by refrigeration and reheating, or by chemical adsorption.

If refrigeration were used the air supply would first be cooled and dehumidified to the required low dew point and then reheated to the required point on the heat ratio line. If a chemical adsorption system such as silica gel were used, the dehumidification of the air would theoretically take place along the line of room wet bulb temperature, say from *A* to *D*, Fig. 5. The air would then be reheated or cooled, as the case might be, along a line of constant dew point temperature to the required point on the heat ratio line. Because of practical considerations in the reactivation of silica gel, the dehumidifying process does not take place along the line of room wet bulb temperature; the line representing the dehumidification process lies above the room wet bulb temperature line. Therefore, when using silica gel, the position of the heat ratio line in relation to the line of room wet bulb temperature is no criterion as to whether it will be necessary to cool or heat the air after it leaves the gel beds.

Notice that Table 3 gives only the angle which the heat ratio line makes with the horizontal line; it does not tell whether the heat ratio line lies to the right or the left of the vertical line of room dry bulb temperature. When the air supply must provide heat for the conditioned space, the heat ratio line must always lie to the right of the vertical line of room dry bulb temperature; when the air supply must remove sensible heat from the conditioned space it must lie to the left. As to the angles, it is necessary to remember only that all of the angles found in Table 3 must lie in either the second or fourth quadrant; *never* in the first or second. On the other hand, the angles in Table 2 must always lie in the first or third quadrant. This fact, together with the rule just stated in regard to the relation between the sensible heat gains or losses and the vertical line of room dry bulb temperature, will instantly locate the correct

Table 1

Conditioned Space	Air Supply	Quadrant in Which Heat Ratio Line Must Lie	Sign to be Used in Calculating Sensible Heat Ratio
Loses sensible heat Loses moisture	Loses sensible heat Loses moisture	1	− −
Gains sensible heat Loses moisture	Gains sensible heat Loses moisture	2	+ −
Gains sensible heat Gains moisture	Gains sensible heat Gains moisture	3	+ +
Loses sensible heat Gains moisture	Loses sensible heat Gains moisture	4	− +

Table 2—Heat Ratio Angles for Various Sensible Heat Ratios. 1st and 3rd Quadrants

Heat Ratio	Heat Ratio Angle	Heat Ratio	Heat Ratio Angle
0.00	90°-0′	0.50	51°-20′
0.02	89°-4′	0.52	49°-5′
0.04	88°-5′	0.54	46°-48′
0.06	87°-5′	0.56	44°-29′
0.08	86°-1′	0.58	42°-9′
0.10	84°-55′	0.60	39°-48′
0.12	83°-46′	0.62	37°-27′
0.14	82°-35′	0.64	35°-7′
0.16	81°-20′	0.66	32°-47′
0.18	80°-2′	0.68	30°-28′
0.20	78°-42′	0.70	28°-11′
0.22	77°-17′	0.72	25°-56′
0.24	75°-49′	0.74	23°-42′
0.26	74°-18′	0.76	21°-33′
0.28	72°-43′	0.78	19°-25′
0.30	71°-4′	0.80	17°-21′
0.32	69°-22′	0.82	15°-21′
0.34	67°-36′	0.84	13°-24′
0.36	65°-46′	0.86	11°-30′
0.38	63°-53′	0.88	9°-41′
0.40	61°-56′	0.90	7°-54′
0.42	59°-55′	0.92	6°-12′
0.44	57°-51′	0.94	4°-34′
0.46	55°-41′	0.96	2°-59′
0.48	53°-33′	0.98	1°-28′
		1.00	0°-0′

Table 3—Heat Ratio Angles for Various Sensible Heat Ratios, 2nd and 4th Quadrants

Heat Ratio	Heat Ratio Angle	Heat Ratio	Heat Ratio Angle
1.00	0°-0′	−3.00	59°-3′
1.10	6°-29′	−2.80	59°-29′
1.20	11°-46′	−2.60	59°-59′
1.30	16°-6′	−2.40	60°-33′
1.40	19°-39′	−2.20	61°-12′
1.50	22°-37′	−2.00	61°-56′
1.60	25°-7′	−1.90	62°-21′
1.70	27°-14′	−1.80	62°-48′
1.80	29°-3′	−1.70	63°-16′
1.90	30°-38′	−1.60	63°-47′
2.00	32°-0′	−1.50	64°-21′
2.20	34°-17′	−1.40	64°-59′
2.40	36°-6′	−1.30	65°-40′
2.60	37°-34′	−1.20	66°-26′
2.80	38°-47′	−1.10	67°-16′
3.00	39°-48′	−1.00	68°-12′
3.50	41°-46′	−0.90	69°-15′
4.00	43°-9′	−0.85	69°-49′
5.00	44°-60′	−0.80	70°-26′
6.00	46°-10′	−0.75	71°-5′
8.00	47°-34′	−0.70	71°-46′
10.00	48°-22′	−0.65	72°-30′
15.00	49°-24′	−0.60	73°-18′
20.00	49°-55′	−0.55	74°-9′
30.00	50°-23′	−0.50	75°-4′
$H_S=H_L$*	51°-20′	−0.45	76°-3′
−30.00	52°-16′	−0.40	77°-7′
−20.00	52°-43′	−0.35	78°-17′
−15.00	53°-8′	−0.30	79°-32′
−10.00	53°-59′	−0.25	80°-55′
−8.00	54°-35′	−0.20	82°-24′
−6.00	55°-34′	−0.15	84°-3′
−5.00	56°-19′	−0.10	85°-50′
−4.00	57°-24′	−0.05	87°-49′
−3.50	58°-7′	−0.00	90°-0′

*When sensible heat equals latent heat, the heat ratio line coincides with the line of room wet bulb temperature.

quadrant in which to draw the heat ratio line. The heat ratio angle is always measured, either up or down, from the horizontal dew point line through the point representing the room condition.

Cooling and Dehumidifying

Problems which fall in the third quadrant have been discussed in detail in previous articles.[2] However, those problems were discussed only from the standpoint of comfort cooling. Problems in process work which fall in the third quadrant are handled in exactly the same manner as for comfort cooling, if the latent heat required for the evaporation of the moisture is not provided by the conditioned air supply itself.

A few words of explanation are needed here. In all of the problems dealing with comfort cooling discussed previously,[2] it has been tacitly assumed that moisture was added to the atmosphere in the form of vapor so that no latent heat was needed to evaporate this moisture into the air. Human beings not only liberate moisture but also provide the latent heat necessary to evaporate this moisture into the air. As a result, moisture from human beings reaches the atmosphere in the form of vapor, not liquid. The same statement is true of such restaurant equipment as a coffee urn, since the moisture issuing from the urn is already in the form of steam, the latent heat required for the evaporation of the steam having been provided directly by the gas burner under the urn.

However, there are many cases in process air conditioning where the latent heat needed for the evaporation of the moisture must be provided by the air supply itself, since there is no other source of latent heat. For instance, in Example 4, it was stated that the wet objects rested on steam coils which supplied the latent heat needed for the vaporization of the water directly to the objects being dried.[3] If no steam coils had been provided, the latent heat needed for the evaporation of the moisture could have come from only one source: the surrounding air in contact with the wet objects. The temperature of the surrounding air does not drop, however, because whatever heat it surrenders is constantly being replenished by the sensible heat gain of the room. In such a case the latent heat needed for the evaporation of the moisture would be provided by the sensible heat gain of the conditioned space.

Example 5—A conditioned room in which solid objects are to be dried has a sensible heat gain of 142,000 Btu per hr. The weight of moisture to be evaporated is 90 lb per hr. There is no direct means, in the room, for providing the heat necessary to vaporize the water. The conditioned space is to be maintained at a dry bulb temperature of 74 F and a relative humidity of 40 per cent. If the dew point temperature of the conditioned air supply is to be 35 F, find the dry bulb temperature required.

Solution—

Latent heat required for evaporation of moisture $= 90 \times 1050$
$= 94{,}500$ Btu

Since the latent heat needed for the evaporation of the

[2] "New Uses for the Psychrometric Chart in Simplifying Air Conditioning Problems," by William Goodman, Heating, Piping and Air Conditioning, Feb., Mar., Apr., and May, 1936.

[3] There is no intention to imply that the wet objects must rest on, or be above, the source which is providing the necessary latent heat for the evaporation of the moisture. If some means is available for circulating air between the objects being dried and the source of heat, there need be no direct physical contact between the source of heat and the wet objects. Unit heaters are often used for this purpose.

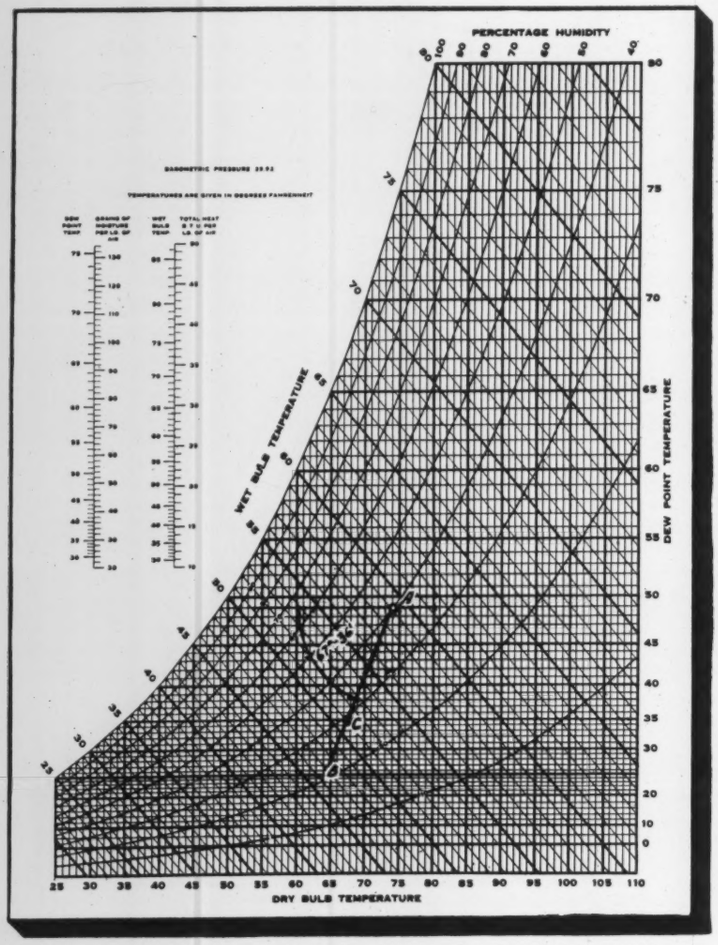


Fig. 6—Solution of Example 5, cooling and dehumidifying

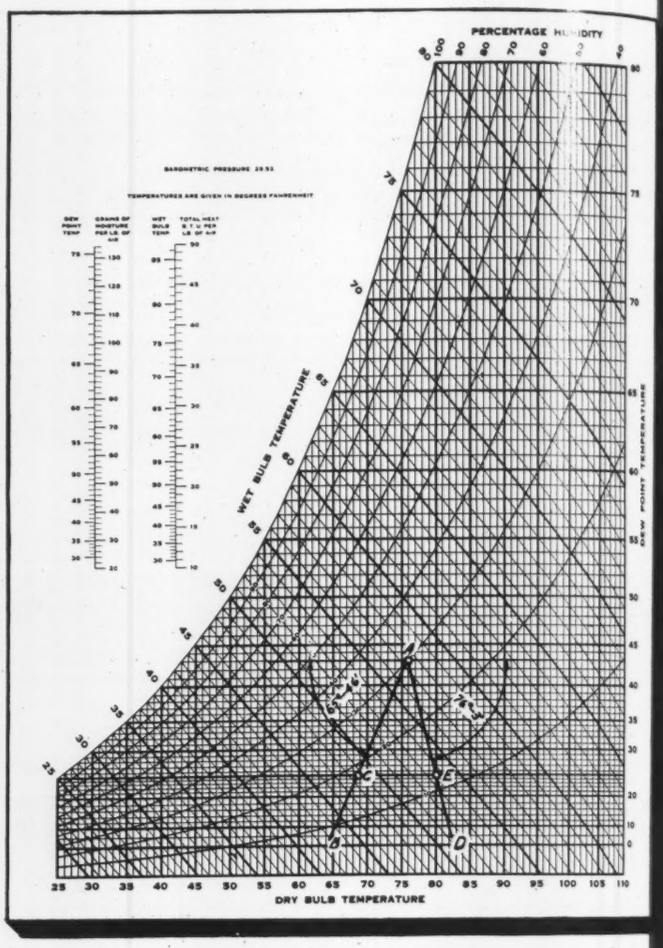


Fig. 7—Solution of Example 6, supplying sensible heat deficiency

moisture comes from the sensible heat gain of the room, the net sensible heat gain which must be absorbed by the conditioned air supply is found as follows:

Net sensible heat gain = 142,000 — 94,500 = 47,500 Btu per hr

$$\text{Sensible heat ratio} = \frac{47{,}500}{47{,}500 + 94{,}500} = \frac{47{,}500}{142{,}000} = 0.335$$

Referring to Table 2,

Heat ratio angle = 67° — 36′

The point *A* of Fig. 6 represents the required room condition. The line *A-B* is the heat ratio line. The dew point temperature line of 35 F crosses the line *A-B* at point *C*. The required dry bulb temperature of the air supply, as read at point *C*, is 67.5 F.

Supplying Latent Heat Deficiency

There are many cases in industrial drying work where the sensible heat gain of the conditioned room is much less than the amount of heat needed to evaporate the required weight of moisture. The deficiency of sensible heat may, in this event, be supplied by a source of sensible heat installed directly in the conditioned space. This procedure amounts to increasing the sensible heat gain of the conditioned room and the problem may be handled in the same way as that of Example 5.

The deficiency of sensible heat needed to evaporate the moisture could also be provided by the air supply itself. This simply means that the dry bulb temperature of the air supply would be higher than would ordinarily be required. As the dry bulb temperature of the air supply falls in the conditioned space, the sensible heat so liberated provides the additional heat needed to evaporate the moisture.

EXAMPLE 6—A drying room is to be maintained at a dry bulb temperature of 76 F and a relative humidity of 30 per cent. The weight of moisture to be evaporated is 128 lb per hr and the sensible heat gain of the conditioned room is 74,000 Btu per hr. The air supply can be provided at a dew point temperature of 25 F. Find the required dry bulb temperature of the air supply if (a) A direct source of heat, such as a unit heater, is installed which will provide sufficient heat to evaporate the moisture; (b) there is no direct source of heat for evaporating the moisture.

(*a*)

Latent heat required to evaporate moisture = 128 × 1050 = 134,300 Btu hr

$$\text{Sensible heat ratio} = \frac{74{,}000}{74{,}000 + 134{,}300} = \frac{74{,}000}{208{,}300} = 0.355$$

Referring to Table 2,

Heat ratio angle = 65° — 46′

The point *A* in Fig. 7 represents the room condition of 76 F dry bulb and 30 per cent relative humidity. The heat ratio line for this case is *A-B*. If the dew point temperature of the air supply is to be 25 F, the required

dry bulb temperature of the air supply as read at the point *C* is 68.8 F.

(*b*) Since there is no direct source of heat for evaporating the moisture, this heat must be obtained from the sensible heat gain of the room. But the sensible heat gain of 74,000 Btu is less than the 134,300 Btu needed to evaporate the moisture. The difference, or 60,300 Btu (134,300—74,000) must be provided by the air supply. This means that the air supply must have a dry bulb temperature higher than the room dry bulb temperature. In cooling to the room dry bulb temperature, the air supply surrenders the additional heat needed to evaporate the moisture.

Since there is no latent heat gain in the room, only water in liquid form to be evaporated, it is evident that the total heat gain of the room is equal to the sensible heat gain through the walls (since the room dry bulb temperature is presumably below the outdoor dry bulb temperature) plus the sensible heat surrendered by the air supply, which amounts to a total of 60,300 + 74,000, or a total heat gain of 134,300 Btu. Since all this total sensible heat is used to evaporate the moisture, the total sensible heat gain is equal to the latent heat gain and also the total heat gain.

The problem is now handled as if the room had a sensible heat loss of 60,300 Btu per hr and a total heat gain of 134,300 Btu per hr.

$$\text{Sensible heat percentage} = \frac{-60{,}300}{134{,}300} = -0.447$$

Referring to Table 3,

$$\text{Heat ratio angle} = 76^\circ - 3'$$

In Fig. 7 the line *A-D* represents the heat ratio line for this case. If the dew point temperature of the air supply is to be 25 F, the required dry bulb temperature of the air supply as read at the point *E* is 80 F.

It is apparent from Example 5 and Part *b* of Example 6 that the total heat gain of rooms in which no direct source of heat is provided for the evaporation of moisture is equal to the sum of all the sensible heat coming into the room, regardless of whether this sensible heat flows in through the walls, is provided by lights, or is released by the air supply. However, when calculating the sensible heat ratio, the total heat gain should be treated as if it were latent heat and its sign should always be positive.

Selecting Condition of Air Supply

In each of the examples which have been presented, the condition of the air supply has been chosen at only one point on the heat ratio line. It is evident that the air supply for each of these examples may be in any condition which can be represented by a point on the heat ratio line in question. The selection of the proper point depends to a great extent upon the character of the apparatus available to condition the air supply. With some combinations of equipment it is possible to obtain air in almost any desired condition. Under these circumstances the selection of the proper point on the heat ratio line rests solely upon the judgment of the designer. Under other circumstances, the latitude of the designer is severely restricted because the available equipment will provide only air having certain definite characteristics as to final humidities or wet bulb temperatures.

[A full page reproduction of the chart used in this article was published on page 88, February, 1936, H. P. & A. C.]

Synchronizing Motor Control—

[*Concluded from page* 371]

potential. This negative potential is impressed upon the grid by a constant d-c source. The electromagnetic generators of the two units are so connected and adjusted that it requires the simultaneous generation of voltage in both inductive generators to overcome the negative bias on the grid of the tube, and cause it to "break down" or "fire" and become conducting.

Since one motor is operating at its synchronous speed and one on its starting winding at its induction or slip speed, it is evident that the voltage peaks from the two generators will coincide sooner or later. When the tube comes conducting, it energizes the high speed relay which applies excitation to the second machine. This relay is relieved of its load and de-energized as soon as the regular control field contactor functions to close.

The two electromagnets are located with the same relative angular displacement as is desired for the compressor cranks. In addition a certain correction is made in this relative angular location to allow for the time lag of the control in order to assure that the motor will pull into step without slipping a pole.

Method Permits of a Number of Applications

This high speed control permits of a number of other very interesting synchronizing applications. The maximum synchronizing torque can always be obtained if the field is energized with the rotor and stator fields at a definite angular relationship to each other. This may, in certain drives, permit using a standard motor rather than one of special design or an oversized standard motor in order to generate the required synchronizing torque. Being able to obtain the maximum torque at synchronizing can also mean lower starting kva for certain types of drives.

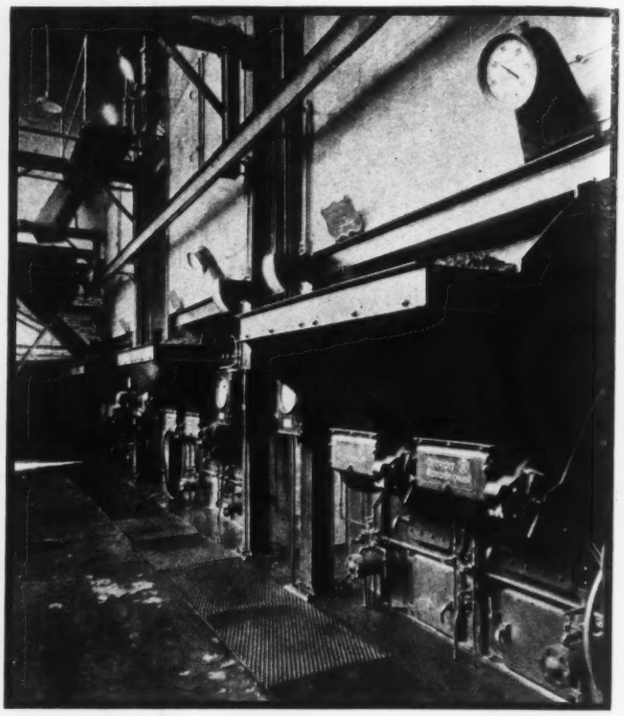
Another application for this control is where a minimum mechanical shock to the apparatus is desired when synchronizing. By applying the field at a pre-determined displacement angle, there is less torque change and, consequently, less tendency for a rotor to over-swing and cause a reversal of the torque transmitted to the drive.

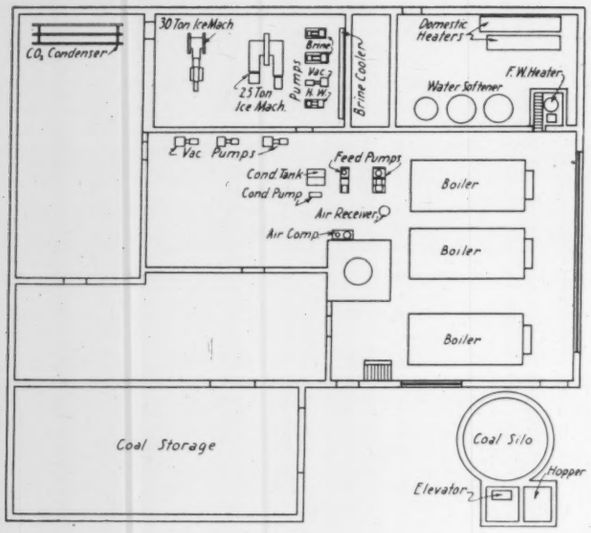
Power Requirements for Air Conditioning

Under this head on page 328 of the June H. P. & A. C., it was stated that the number of kilowatt hours per season per compressor horsepower is about 100 for an office building. This is obviously in error; the correct figure is 1000.

Controlling Construction Costs

By C. C. Wilcox*





Firing aisle, showing stokers and weigh larry
At left—Plan view of the central steam plant

Committment Method, Weekly Analyses, Enable Control of Expenditures for Constructing Central Heating Plant at St. Mary's Academy

ACTUAL construction costs frequently exceed estimated costs by a considerable amount, a condition usually inexcusable and always embarrassing to designer, constructor, and owner. A knowledge of accumulated cost totals day by day is helpful in giving proper control of these costs as the work proceeds.

Of course, careful estimates must in any event precede actual spending of money, but even with such estimates this budgeted total may be exceeded unless accumulated total expenditures are closely watched. These may be kept either on the basis of bills paid or committments made, the latter method having the advantage over the former of presenting at the end of each day the accumulated total to that time.

This committment method was used to exercise control over the construction expenditures during the building of the steam plant for St. Mary's Academy near South Bend, Ind., which serves buildings having a total of 125,900 sq ft of equivalent direct radiation, 60,535 sq ft of glass area, 269,133 sq ft of 12 in. brick wall area, 165,019 sq ft of roof, and 7,691,290 cu ft of contents.

The plant was completed in January, 1935. Every committment for expense, either in labor or material, was kept in detail each day, and once a week this accumulated total was added to the estimated balance necessary to complete the plant and so compared with the estimated cost of construction. Estimated cost was $65,569 and the actual expenditures $64,160. In addition, $3,912 of work was done which was not contemplated in the estimate, but which is included in the analysis on the next page.

Of course the old plant had to be kept in operation during construction of the new and both construction costs and operating costs were therefore undoubtedly increased somewhat. The nearest railroad siding is about 4000 ft from plant and all material had to be hauled that distance. The water softeners, the feed water heater, and the 25 ton ice machine were all in place when work started, and are not included in the costs shown. The domestic water heaters, one brine pump, one vacuum pump, and the hot water circulating pump were on hand but the cost of setting and piping is included in the figures given.

*Mechanical Engineer, South Bend, Ind.

Analysis of Cost and Man-Hours of Labor for Central Steam Plant, St. Mary's Academy

Item	Cost, in Dollars				Man-Hours of Labor											
	Total	Contract	Other Material Furnished by Owner	Labor	Total	Electrician	Elect. Helper	Steam Fitter	Steam Fitter and Welder Helpers	Welder	Brick Mason	Mason Helper	Labor	Pipe Coverer	Pipe Coverer Helper	Machinist
Building and wiring	$ 6664	$ 4928	$ 920	$ 816	1336	757	244	19	14	42			260			
Remove old boilers	405		165	240	528					115			413			
Boilers (steel)	13730	13730														
Boiler settings	9159		5841	3318	5347			99		74	2564	1142	1468			
Piping	6272		3001	3271	4339			988	1465	1713			173			
Pipe covering	1621		656	965	1477									1185	292	
Stokers	7654	7591		63	136			27	27				82			
Walkway grating	245		122	123	260			71	107	72			10			
Coal and ash handling	6371	6226	45	100	218			9	47	20			142			
Combustion control	589	575	6	8	14			6	6				2			
Soot blowers	1076	770	152	154	285			82	102	95			6			
Boiler feed pump	266		145	121	205			25	49	24			58			49
Stoker fans and ducts	1772	1611	109	52	96			4	43	49						
Breeching & chimney open.	625		334	291	492			119	209	94	31		39			
Feed water regulators	416		390	26	48			19	18	11						
Ice machine	6147	5450	271	426	837			74	225	127			411			
Deep well pump	1542	1088	255	199	407			28	77	98			204			
Brine pumps and cooler	863	359	241	263	498			32	244	186			36			
Air compressors	137		86	51	105			16	48	25			16			
Meters	2518		2409	109	209			107	96	4			2			
Total	$68072	$42328	$15148	$10596	16837	757	244	1725	2777	2749	2595	1142	3322	1185	292	49

Credit extras not originally contemplated.. 3912

$64,160

Building—The original building, a single story brick structure, was remodeled to house the new equipment. That portion of the building which was rebuilt measures 37 ft x 55 ft in plan and 30 ft from floor to bottom of flat roof trusses. The entire roof, the upper third of three walls, the steel sash, and floor are new. There is no basement. The rest of the building as shown on accompanying plan was not changed.

Coal Handling—A concrete silo 16 ft-0 in. diameter, 47 ft-6 in. high, and 4 in. thick with upper shelf for 38 tons live storage, and space below for 169 tons mechanically reclaimed storage, 20 ton per hr bucket elevator, concrete unloading hopper, and 2000 lb weigh larry, comprise the coal handling equipment.

Boilers—Three 300 hp, 200 lb low head bent tube boilers were installed, set singly. Boilers were set to give a furnace volume of 1056 cu ft each. Material requirements consisted of 25,870 No. 1 fire brick, 20,500 No. 2 fire brick, 25,200 common vitreous twins, 302 sacks of prepared mortar, and 18,825 lb of high temperature cement for the three settings above foundation line.

Removal of Old Boilers—The old boilers consisted of six H. R. T. boilers 72 in. x 18 ft and two Scotch marine which were cut up for junk. In the cost of removing the old boilers was included the moving of the domestic water heaters to the new location in the softener room. This account was not credited with amounts realized from salvage of old boilers.

Piping—Main and auxiliary steam headers were installed and duplicate feed water headers to points opposite each boiler. The equipment piped up consisted of three boilers, two feed pumps, one air compressor, two brine pumps, one vacuum pump, live and exhaust steam connections to outgoing lines, one 30 ton CO_2 refrigerating compressor, condenser and brine cooler, two domestic water heaters, one deep well pump, and combustion control. Practically all piping 2 in. and above was acetylene welded, and on the larger sizes welding fittings were used. In order to save time, the feed water headers and as much of the steam and other piping as was found practical were weld fabricated on the ground outside before boilers were placed and then brought into the plant for assembly. This facilitated welding and undoubtedly reduced the man-hours required. The two pipe condenser which had been used with an abandoned CO_2 compressor was rebuilt by welding to serve as a direct expansion brine cooler for the new layout. Exhaust steam connections from CO_2 compressors, vacuum pumps, and feed pumps were carried to feed water heater already installed.

The deep well pump was installed on an existing well casing about 100 ft from the old pump house, to which the discharge had to be run and connected. Blow-off connections were carried to new tank buried in ground just outside building wall, and vented to roof. None of the old valves and very little of the old piping could be used in the new plant although many of the larger valves required in the new plant were rebuilt used valves of required grade, whose service history was known.

Pipe Covering—Used pipe covering constituted about one-third of the total, and the balance was new. It is all 85 per cent magnesia of 1 and 2 in. thickness as required, except for cork on the brine lines.

Stokers—Double rotor overfeed stokers, having 65 sq ft of grate area were installed. They are equipped with air operated dumps and undergrate blast fan, turbine driven. Stoker rotors are arranged for individual dual drive.

Walkway Grating—Approximately 200 sq ft of so called "subway" grating was installed to provide access to drum heads, water columns, etc. Used grating was available at about 25c per sq ft.

Combustion Control—Water operated pressure control was provided to regulate coal feed and air intake to fan. The master control operates a rotating shaft serving each boiler from which connection was made to stoker, to fan, and later to uptake damper.

Soot Blowers—Three soot blowers were installed in each boiler, one calorized and two plain elements.

Boiler Feed Pumps—One existing simplex boiler feed pump was reset and a used 10 x 6 x 10 duplex pump was purchased and installed as a spare.

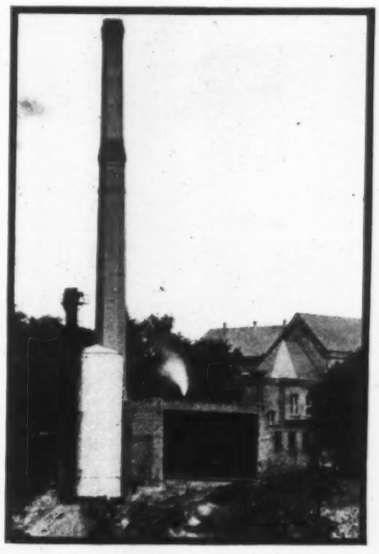
Breeching—No. 10 gauge electric welded steel breeching was installed on supports from rear of boiler setting and connected into a new opening provided in the brick stack. The breeching was not insulated. The sheets were cut to shape in the shop and assembled by electric welding in place.

Feed Water Regulators—Feed water regulators of the generator type and control valves were installed.

Ice Machine—A 30 ton steam driven CO_2 ice machine with two pipe condenser and rebuilt direct expansion brine cooler was installed. To this account was charged the removal of the old ice machine, but no credit allowed for its salvage.

Air Compressor—A used two stage direct steam driven compressor was provided for operation of stoker dumps. A 24 in. x 10 ft air receiver was installed with it.

Meters—Boiler meters and three point draft gauges were provided for each boiler.



New central heating plant. (View taken with telephoto attachment)

Underground Piping for Central Refrigeration Service

By R. H. Switzler*

Four U. S. cities are served by central plant refrigeration, two of the systems employing brine circulation and two direct expansion. In point of area covered, the St. Louis system, a franchise for which was originally granted in 1890, is believed to be the largest. . . . Details of the piping construction used and the solution of the operating problems which have been encountered were described at the International Congress of Refrigeration held last month

DEVELOPMENT of refrigeration into systems of underground refrigerating pipe lines served by central station plants finds expression in four U. S. cities, Boston, New York, Los Angeles and St. Louis. The pipe lines in Boston and New York furnish brine service, cold brine being circulated in underground mains from a central plant; in St. Louis and Los Angeles service is by direct expansion, ammonia being circulated from the central plant to the customer's premises where the refrigerating work is done by the expansion of the liquid ammonia into the customer's apparatus.

In point of refrigeration delivered to customers, measured by the revenue produced, the system in Boston is probably the most important; however, in the point of physical area covered, the St. Louis system is believed to be the largest in existence, with excellent possibilities for future development in view of the modern trend toward air conditioning. This system is owned and operated by the St. Louis Refrigerating & Cold Storage Co., and is the subject of the present paper. The St. Louis pipe line originated in 1890 under a franchise granted by the city. The area covered comprises approximately 25 city blocks north to south and 25 city blocks east to west. A total of about 8 miles of piping is operated, and the distance from the main plant to the farthest point served is about 2½ miles.

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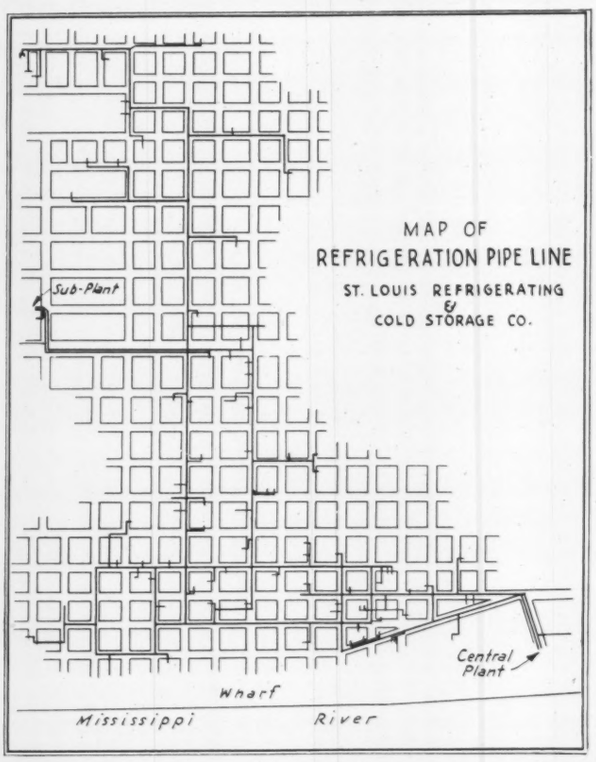
From a paper prepared for presentation at the International Congress of Refrigeration, The Hague, June, 1936.

The operation of this pipe line has entailed much pioneering work, and numerous theories heretofore entertained by refrigerating engineers have been shown to be fallacious. In the development of the original pipe line, problem after problem was encountered with numerous unforeseen occurrences. Present day methods of construction were unknown and the handicaps with the materials available were many.

One of the most important features of construction is that the lines are not laid directly in the earth but are placed in a vitrified tile duct which makes possible a very simple construction in the streets. The original pipe lines were of no great length and some of the troubles encountered did not foretell the seriousness of some of the difficulties to be met in the construction of lines of greater length. For example, it was not realized that lines of great length would increase in expansion as many times as the pipe was lengthened; that is, in a short run of pipe the expansion and contraction is nominal and of little import but when these lines are lengthened considerably this expansion and contraction of the pipe becomes an important factor. At times these lines would pull apart with the resulting loss of a large quantity of ammonia. This problem has been solved by the use of expansion joints, each situated so as to take care of at least 3 in. of travel in every 300 to 500 ft of pipe.

Another vital change made in the construction of these lines is that of welding all pipe joints with the exception of fit-



tings and cross-connections which are situated in manholes constructed at from 300 to 500 ft apart. Before welding, leaks were constantly occurring which not only entailed loss of ammonia but excessive maintenance and reconstruction costs. When a new customer was connected to the line it was necessary to run a new line from the closest manhole to his premises; by means of welding, it is possible to connect a lateral directly into the main line at the closest point available. This lateral is extended to the curb line in front of the customer's premises and stop valves are placed in a small manhole constructed in the sidewalk. These valves are used for shutting off the service to the customer whenever the occasion arises, either through normal operation or through emergency.

One of the lines installed is the vacuum line by means of which it is possible to remove all of the ammonia, whether liquid or gas, from any section of the mains or from the customer's premises. This affords a simple means for making connections or changes and for handling any emergency. This vacuum line is cross-connected into the gas and liquid lines at all manholes and stop boxes.

The pipe lines, placed in a tile duct, are supported on rollers set at every 10 ft. The hangers which carry the rollers are constructed so as to permit such water as may enter the duct to drain to a manhole in which are located drains to a nearby sewer. In the tile duct three lines are placed; the liquid anhydrous ammonia line feeding the customer's apparatus, where the ammonia is expanded through automatic valves; the gas line returning the gas to the plant, and the vacuum line. In some instances a second gas line is added, making it possible to operate at different back pressures.

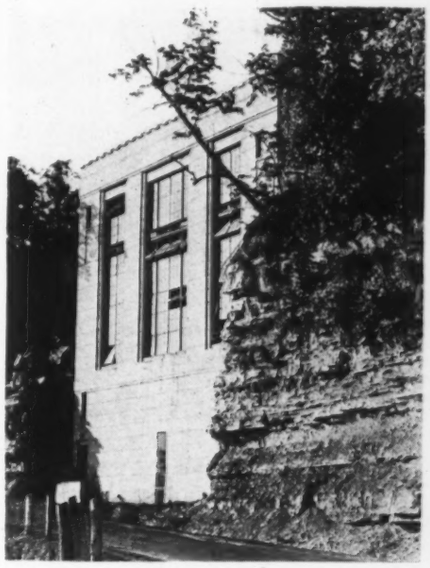
The loss in transmission of refrigeration in the liquid and gas lines is practically zero as the liquid and gas remain at ground temperature. Consequently, no insulation is used on these lines. Latent heat of the gas is removed at the main plant by changing this gas into a liquid and in turn this liquid is transmitted through the pipe line to the customer's premises, there collecting the heat required for the gassification of the liquid ammonia. To state this cycle simply, pipe line operation is the acceptance of the ammonia gas into the central station plant where the latent heat is removed and the gas changed back into a liquid which is then circulated through the pipe lines to the points to be served.

Operating Problems Encountered

Among the difficulties encountered are the problem of super-heated gas returning to the plant and troubles with oil entering into the ammonia. Abnormal pressures are encountered at times due to the pipe lines passing through heated zones in the ground, caused by underground steam lines, or due to the pipe lines having been placed too close to the surface of the street—sometimes unavoidable due to underground obstruction. This problem of gassification due to abnormal heat is met through correction of pressures or by sub-cooling the liquid.

Another problem is that the lines ultimately reach their limits in capacity. This condition was first remedied by adding more lines and increasing their size; later on, by installing a booster plant near the central section of the total line. This sub plant relieved this condition and made it possible to handle more than twice the previous load.

So that it would not mar the beauty of the grounds, the steam plant serving a 300 bed hospital at Hamilton, Ontario, Canada, was constructed in an excavation in the side of the mountain immediately below the hospital. The view shows the side of the building facing the highway which winds up the mountain.



Coal is delivered at the top of the building, level with the ground surface, and is fed by gravity to the stokers. Ashes are sent out the bottom of the building and are carted away.

A hot water heating system is employed. There are two converters for space heating requirements and two for domestic service.—Andrew C. Martin.

The most recent problem of moment has been the rapidly increasing air conditioning load; to meet it with greatest economy in installing underground mains to the sub plant an extra gas line was run along with the customary liquid and gas lines, in order to handle two different back pressures, one for the regular work and one for the air conditioning loads. This has been an important development of the sub plant, namely, the maintenance of a higher back pressure resulting in a reduction in cost of operation. It must be remembered that the cost of manufacturing refrigeration and distributing it through underground refrigerating pipe lines must be kept at a minimum in order successfully to cope with the competition of isolated compressor operation. This is axiomatic of all central station work.

A direct expansion pipe line is believed to have certain advantages over a brine circulating system, not only in the larger area that can be covered but also in the fact that refrigerating losses due to temperature difference, which can be corrected only by costly insulation in the case of a brine system, are not present. Furthermore, the size of lines in a brine system must be much larger than the direct expansion lines for the same load. Furthermore, with direct expansion the service that can be rendered to the customer is more flexible; when the customer desires to circulate brine on his premises instead of using direct expansion this brine is cooled in a brine cooler into which the liquid ammonia is expanded.

PRACTICAL HEATING

By Harold J. Taylor*

Notes on Vacuum and Condensation Pumps

In my experience, I have found heating jobs with pumps incorrectly connected, not properly selected for the requirements of the job, and in many cases involving the use of a vacuum-condensation pump, controls not in good working order. Frequently, the operator has had insufficient information regarding pump maintenance or operation, and but a vague idea of the pump's function.

Use of Pumps

Condensation and condensation-vacuum pumps are primarily installed on a heating system to return condensation to the boiler when the pressure (no pressure return) in the return mains is much less than the boiler pressure and the condensation cannot flow to the boiler by gravity. Condensation pumps are used, as a rule, only as condensation pumps and are usually installed on an open system. Condensation-vacuum pumps are used for creating a vacuum on the return lines as well as returning condensation, and are used on a closed vacuum type of installation.

The condensation pump may be used on jobs where radiation is installed below the water line of the boiler, provided the inlet of the tank receiver is below the outlet of the radiators. A distance of 12 in. should be allowed between the tank inlet connection and the bottom of the lowest radiators; this will allow for the pressure drop in the piping and the necessary head to cause the condensation to flow to the receiver tank. If the return piping to the pump is water pocketed (that is, of a wet return nature), then it is necessary to vent the return lines at the point where they drop to the wet return. Ordinarily, all air from the system is vented through the receiver tank.

A condensation-vacuum pump is used on a vacuum type job. Where lift fittings are used—as where radiation is installed below the pump receiver inlet—a condensation-vacuum pump must be used. With an installation of this kind, the float control feature of the pump cannot be used.

As smaller piping sizes may be employed, it is customary to use condensation-vacuum pumps on large jobs, with a saving in installation and operating cost of the heating system.

A condensation-vacuum pump should not be installed on an existing job unless all traps, radiator valves and piping connections are positively vacuum tight. The pump will not be satisfactory unless a normal vacuum can be maintained with minimum power consumption. I have seen condensation-vacuum pumps installed on existing jobs and the trap elements collapsed when the pump created its normal vacuum setting. I suggest on existing jobs the installation of a condensation pump in preference to the vacuum type, unless it is reasonably certain that the job warrants the vacuum pump. In many cases the owner may be willing to have the heating system repaired to the extent that the condensation-vacuum pump equipment will be ideal.

*Contracting Engineer, Detroit, Mich.

It is important that all radiator traps, drip traps, etc., be in good working order so that they do not pass steam into the return lines to the pump, or the pump will not work satisfactorily. To test a job for leaky traps, open the return lines at convenient locations and subject the system to approximately 10 lb steam pressure; if steam is delivered from one or more of these return lines, it is necessary to repair the defective traps. As a rule, only the trap inside elements and seats will have to be replaced.

It is general practice for pump manufacturers to furnish a float control switch (known as night switch) which is connected only with the vacuum feature of the pump. This switch may be used for night operation and during early fall and late spring when a light fire is carried on the boiler. It is evident that with a light fire, the traps will not close because insufficient steam is available and the pump will run continuously in its endeavor to maintain a normal vacuum in the heating system. This will cause unnecessary pump wear and power consumption.

If the float switch is used, the condensate must be returned to the receiver tank by gravity and from the tank to the boiler by pump. However, if the condensate is not fed by gravity to the receiver tank—that is, if the pump receiver inlet is above the return line from the heating system—the vacuum pump must be used at all times and the float control switch should be fixed in such a way that it cannot be tampered with.

It is only in rare instances that radiation or return piping is below the pump receiver tank inlet; in this event, an accumulator tank may be used, and then the night switch can be operated.

Correcting Pump Troubles

The following suggestions will be helpful in locating and correcting pump troubles:

Is capacity of pump in sq ft of radiation or in pounds of condensate ample for the job?

Is pump capable of pumping condensate to boiler at maximum boiler pressure? If not, and the boiler pressure can be set lower, then replace the pop safety valve with one of a lower setting to suit the pump discharge pressure.

If boiler pressure cannot be changed, then inquire of pump manufacturer as to whether or not a larger motor can be installed, thereby increasing the pump discharge pressure to meet maximum boiler pressure.

Are air leaks present? If so, correct them.

If a duplex type water feeder is used on the boiler, is there a check valve on the feeder over-flow line? If not, install one in a horizontal position to prevent vacuum loss at this point.

A gate valve and 45 deg swing check valve must be installed in the discharge line from the pump to the boiler. This line should be connected to the return header of the boiler through a Hartford loop connection.

Receiver Tank and Pump

Pump receiver must be properly vented to the atmosphere.

Steam must not be present in the receiver tank.

If receiver tank overflows (capacity being ample), pump is not pumping condensation to the boiler. Check the float in receiver tank for water-logged condition by raising float rod

connection. If this causes the pump to operate, the float is either water-logged or has been disconnected from the rod.

I have found a few jobs where the tank would overflow and after inspection, discovered that the steam boiler was priming a quantity of water (due to dirty water) and the pump was not able to carry the additional load. To correct this condition, thoroughly clean or blow off the boiler until the priming ceases.

Check rotation of the motor or pump. Usually an arrow indicates the proper direction of rotation.

Be sure the contact points on the float switch are making contact; if not, switch must be repaired.

With motor running, does the shaft between pump and motor turn? If not, there probably is an obstruction in the impeller, burned out bushing, frozen packing or pump and motor are out of alignment. With motor turned off, the shaft should turn freely by hand.

Is pump primed with water? It should be.

With some types of pumps, it is possible that the discharge line from pump to boiler may be out of line, subjecting the pump to strain and causing it to slow up or even stop the motor. Be sure there is no strain on the pump head due to insufficient support of the discharge line to boiler.

Be sure that the strainers are clean.

Check all valves for their open or closed position and be sure they are correct.

Motor

If motor fails to run, check the following:

Does electricity supplied to motor correspond to specifications on motor plate?

If motor burns out fuses, are they of ample amperage?

Is there a short or ground in the motor?

Are oil and grease cups properly lubricated?

Is commutator clean? If not, sand *lightly* with No. 00 sand paper.

Are air gaps in armature clean? If not, clean with air pressure (if available) or clean between segments with a blunt, soft, wooden stick.

Are brushes sparking? If so, either replace them or point up the old ones. Cleaning of commutator may clear up this sparking condition.

Is motor running at proper speed? This speed may be ascertained with the aid of a speed indicator or tachometer. (This instrument is usually obtainable from an electrician or public utilities company on a loan basis for a short period of time.)

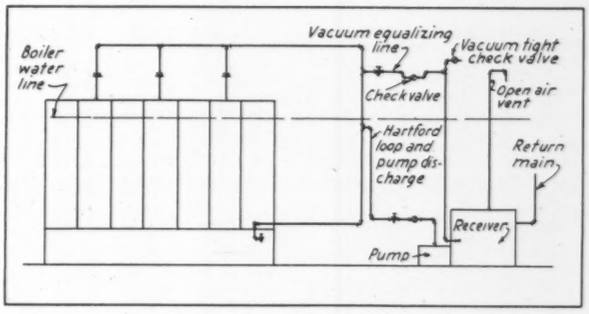
Installing Vacuum Equalizing Line

It is advisable to install a vacuum equalizing line with a two pipe vapor-vacuum heating system to safeguard the steam boiler from low water caused by an unequal vacuum forming between the steam supply and dry (no pressure) return lines. This equalizing line allows for equalization of vacuum in the system if a higher vacuum tends to form in the steam supply piping than in the return piping. A high vacuum in the supply line results when the fire is banked, or when it is shut down on mechanically fired jobs.

Where a vacuum pump is installed and is at times operated on float control, the probable forming of a higher vacuum in the steam supply line will prevent the water of condensation from flowing to the pump by gravity and boiler breakage may occur. When a condensation pump is installed on a vacuum type job, the same boiler low water condition may prevail.

The ¾ in. vacuum equalizing line is installed as follows—

On a vacuum pump job, a vertical riser is run from pump outlet (provided for this purpose by pump manufacturer) to a point above boiler water line level. At high point, install a vacuum tight horizontal check valve opening toward atmosphere.



Method of connecting vacuum equalizing line when vacuum pump is used and return to pump is above receiver tank inlet

In a tee connection below check valve, extend a horizontal pipe line to steam boiler header or, if more convenient, connect to any opening above the water line, such as below the pop safety valve or draft regulator opening. In this line install a horizontal swing check valve (opening toward boiler) in a water trapped position. In other words, the check valve should be placed below the horizontal run of piping by using four ells and four nipples. This connection is made at a convenient location in the boiler room.

When the condensation is returned by gravity to the pump, that is, when return is above inlet of receiver, the vacuum equalizing line is connected between the boiler header and the outlet on the pump.

When condensation is returned to the pump through the use of a lift fitting, that is, when return line is below inlet of receiver and the distance does not exceed 30 ft between the pump and point where the dry return drops below boiler water line level, install the equalizing line as suggested in the above paragraph. When the distance exceeds 30 ft, a ¼ in. brass pipe vacuum regulator line should be installed—in addition to the equalizing line—from the pump outlet provided by the manufacturer to a point half-way between the pump and where the dry return drops below boiler water line level.

When an accumulator tank is installed in connection with a vacuum pump, the vacuum equalizing line is connected between boiler header and top connection of tank. In addition to the vacuum equalizing line, a ¼ in. brass pipe vacuum regulator line is connected between a tee connection above tank (in vertical riser of vacuum equalizing line) and outlet on pump provided by pump manufacturer.

When a condensation pump is installed on a vacuum type job, extend air vent line from pump receiver to a point above water line of steam boiler and install a vacuum type quick vent valve. At a point just below quick vent valve, install a tee connection and connect ¾ in. vacuum equalizing line to header of steam boiler.

Note—It is advisable to consult the manufacturer of the pump or accumulator tank in regard to connection details pertaining to a specific make of equipment.

A good 45 deg check valve should be used on this type of installation; I have found two jobs equipped with vacuum equalizing lines where it was difficult to create a vacuum and after a thorough inspection, I found that the check valve was not holding tight and steam was allowed to pass into the pump from the steam boiler header.

How to Make a Pipe Templet

By S. C. Clark*

DURING the installation of a 200 ft, 20 in. diameter, oxy-acetylene welded pipe line, which connected a coal mine's new steam turbine with the spray pond, several changes in direction were required, each of which necessitated the laying out of a miter joint and subsequent position welding. The method used is described in the following paragraphs:

A simple plan of the intersection was all that was necessary to develop the templet. With the angle of the turn known, two straight lines were laid out to represent the center lines of the intersecting pipe. This is shown in Sketch 1. These two lines are represented as *AB* and *BC*, intersecting at Point *B*.

The angle formed by the intersection of these two lines was then bisected. Where a protractor is not available, this can be done by taking points such as *A* and *C* equidistant from the intersection of the lines and marking off arcs of circles, using *A* and *C* as centers. Where these two equal arcs cross, a line is drawn through this point *E* to the intersection *B* of the angle. This is an exact bisection. This line also represents the angle for the miter cut.

Next, as shown in Sketch 2, a half circumference of a circle equal to the diameter of the pipe was drawn with a point on line *AB* as center. Then a number of equally spaced points were marked off on this circumference. Lines parallel to the center line were then drawn from these points through to the miter cut line, EE'. Next, from that point which on the drawing represents the inner edge of the miter joint, a line known as the "base line," was drawn perpendicular to the center line. Then the distance from this base line to the line *EE'*, forming the miter joint gave the determining lengths for laying out the templet. These distances have been marked on the sketch as *a, b, c, d, e, f* and *g*.

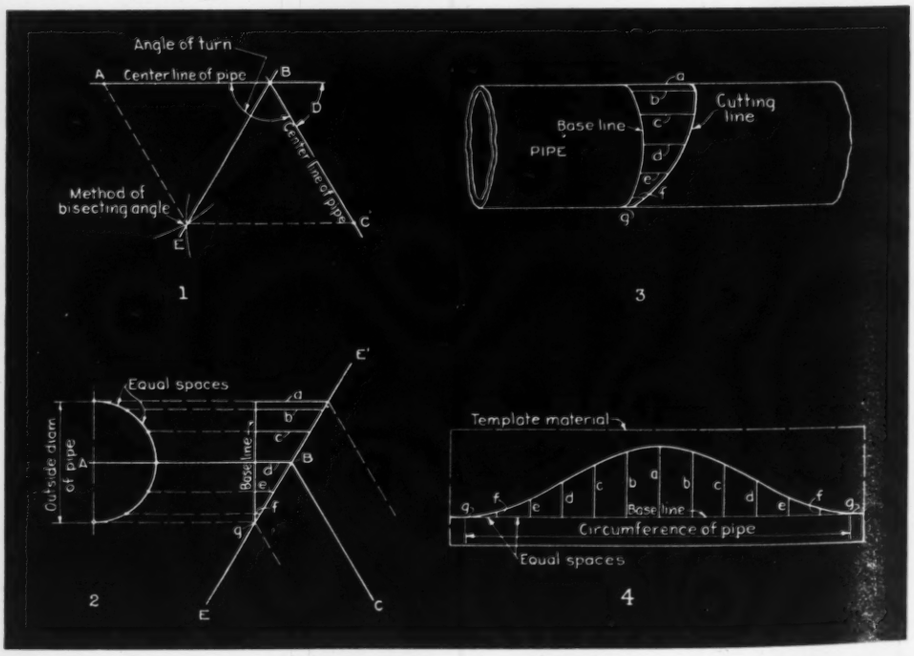
With these distances determined, there are two ways in which the operator could have developed his work. He could have drawn a base line around the pipe with chalk at the point where the turn was to be, then marked off on this line the required number of points equally spaced; and from these points, parallel with the center line of the pipe, distances measured off corresponding to *a, b, c, d,* and so forth, on the pipe, and drawn a line through the ends of these to guide his cut. This is shown in Sketch 3. Where only a single turn is to be made, this is the simplest procedure.

It is important here to draw attention to the fact that the distances between the horizontal projection lines shown in Sketch 2 are not the actual spacing of these lines as they are on the pipe or templet. It is evident that it is the points from which these horizontal lines are drawn on the semi-circle that represents the pipe circumference which are spaced apart equally.

In this case, there were other joints of the same angle to be made, and so it was advisable to make a regular templet on heavy paper, which could be retained for further use. Thin cardboard or sheet metal also makes good templet material.

The templet material was first cut so that it was slightly longer and wider than the finished templet was to be. This permitted making overlaps on the templet at each end in order to secure the templet in place on the pipe when marking.

The marking of the templet was similar to the method just described of marking the pipe directly. A base line was laid out on the templet corresponding in length to the entire circumference of the pipe. This base line was then divided into twice the number of equal spaces as were used on the half circumference used in Sketch 2,



(1) Rough sketch illustrating the first basic lines necessary to lay out a templet for pipe turns. (2) The second step in laying out a templet. (3) From sketch 2, the pipe itself can be marked if no further templet is desired. (4) Where other joints of the same angle are to be made, a permanent templet is useful

*The Linde Air Products Co., New York.

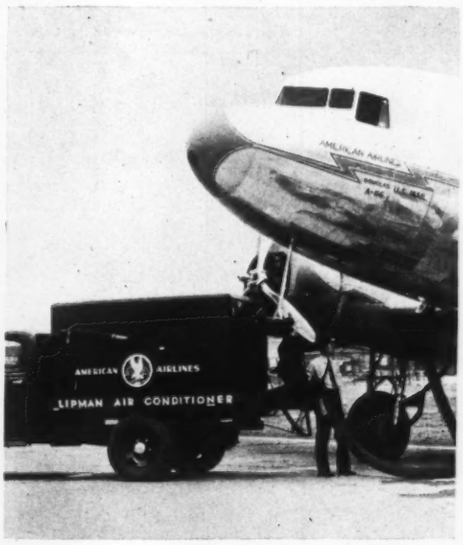
and lines perpendicular to the base line were drawn from each point. Using dividers, calipers, or simply a measuring stick, the distances on the plan corresponding to the length of the lines *a, b, c,* and so forth, were carried on to the templet, and positioned on the corresponding perpendicular lines. Then through the outer ends of these measuring lines a smooth curve was drawn and the templet cut out ready for use.

Once the chalk mark was made on the pipe, using the templet as a guide, it was found to be preferable to center-punch the line at several points on the pipe in order to facilitate accurate cutting.

In making the actual cut on the pipe, the blowpipe flame was held in the plane of the cut, in such position that a true miter cut was produced. When this cut was completed, the edges were beveled for welding. In order to aid accurate alignment, the bevel was extended only to within 1/32 in. of the inner pipe wall.

When the cuts had been made, the joint was lined up by rotating one of the sections of pipe 180 deg. Then the spacing of the joint was fixed so that there was slightly more space at the inside of the turn than at the outside. This allowed for contraction in the weld, which is invariably greater on the inside of the turn. Then, after the joint was tack-welded at the quarter point, the angle of the turn was found to be slightly greater than that desired. The approximate degree of necessary increase had to be worked out by a trial or two, but the completion of one or two turns gave the operator a good idea of what this had to be.

Portable Units Condition Air Liners



WHILE in flight, planes are heated in winter from the engine exhaust, and a ventilating system maintains comfortable cabin temperatures in summer. However, while in the air port, without the engines running, the cabins cool rapidly in winter and in the summer will quickly warm up unless cool air is supplied to the ventilating system.

Developed to overcome this problem, the unit shown in the left view cooling the American Airlines' giant new flagship has a lower section consisting of an insulated hold-over tank containing steam coils or electrical resistance for heating the tank contents for winter heating service, and icing coils for cooling service. In the summer, water is used in the tank. When heating is required, an anti-freeze solution is used. The liquid is circulated through the coils of a unit heat exchanger and filtered air passes around these coils. A fan delivers the conditioned air through a 5 in. flexible tube directly to the plane body or to the inlet of the plane ventilating system.

A "Freon" compressor is connected to the coils so that a mass of ice can be frozen in the tank; this machine is run at any time when the unit is not being used for cooling purposes.

Illustrated at the right is a portable cooling or heating unit of a type used by Transcontinental & Western Air, and by United Air Lines. Precooled or preheated calcium chloride brine is employed, and the unit has capacity to introduce 1200 cfm of filtered, cooled and dehumified air to the cabin.

Boulder Dam Engineers Solve Cooling, Ventilating Problems

Air Circulation System Maintains Constant Penstock Temperatures; Power House Offices, Control Rooms, Cooled • *By* Carleton W. Campbell*

Cooling and ventilation problems at Boulder Dam have been handled in a most interesting manner . . . Briefly described here is the system of air circulation for the penstock tunnels, where temperature must not vary too greatly in order to avoid trouble from expansion. Ventilating and cooling the power house proper—equivalent to a twelve story building five blocks long—is also explained

THE Boulder Dam power house is about equal in size to a twelve-story building five city blocks long and is located at the base of the dam in the Colorado river canyon, 1200 ft below the general level of the ground surface. It is therefore surrounded on three sides by high walls and as this is a very hot part of the country in summer air cooling and ventilation were important problems which have been handled in a most interesting way.

Controlling Temperature of Penstocks

There are two entirely separate systems of air circulation involved, one to serve the main penstock tunnels and the other for the power house. There are two main penstock tunnels 37 ft in diameter and two 50 ft in diameter each housing a 30 ft header that has no expansion joints—which were not provided because of the great size of the penstocks and the 225 lb per sq in pressures involved. Since these giant tubes are each several hundred feet long the thermal expansion would be serious; therefore the temperature must not change much.

Of course, when water is flowing the temperature will remain fairly constant between limits of about 50 and 60 F, but for installation the sections have to be cooled and kept at about the normal working temperature until put into service; furthermore, whenever one is out of service for repairs the temperature must not vary greatly from the mean of 55 F. Each tunnel is therefore provided with a ventilating conduit that reaches to the upper dead ends and is connected to a 24,000 cfm centrifugal exhauster. Buried in the 2¾ in. steel walls of the penstocks, thermostats control the opening and closing of automatic doors and the operation of the blowers.

For installation purposes 30 ft lengths of the penstock are taken in through the adits (see sketch) and joined, leaving a temporary girth joint open about every 200 ft. Then the final joining of these sections is accomplished after the metal cools to the approximate working temperature, the cooling being accomplished either by taking in outside air at natural temperatures during the cooler months or by refrigeration, for which a 900 ton plant is provided. During operation no temperature variation can occur. In the event of a shutdown for repairs, men will be sent into the tunnels and air supplied them through the adits, the air being drawn up the tunnel along the penstocks and exhausted back through the ventilating pipe for one complete change in 58 minutes.

If the temperature of the outside air is close to the permissible temperature, then it can be circulated freely without danger. If the air is cold and the temperature of the penstock falls, the thermostats in the penstocks will cut down the air intake or cut it off entirely. The same will occur if the air warms up the penstock to the predetermined danger point. But since the mass involved is great and the temperature variation permissible in the penstock is around 30 F most of the control lies in the *volume* of air circulated. For emergency purposes, a spray screen across the mouth of the adit tunnels can be used to reduce the temperature of the incoming air in the hottest weather quickly and since the tunnel walls are constantly warm they will assure plenty of warmth in the coldest weather when the incoming air volume is controlled.

Ventilation of the Power House

The plan for ventilating the power house is as follows: Air is taken through the adit tunnels into each of the four main penstock tunnels where it divides and passes through the sixteen branch tunnels along the

*Mechanical and Industrial Engineer, Denver, Colo.

penstocks into the power house. Each main tunnel has four branches and there is an upper and a lower tunnel on each side. The branches from the upper and lower tunnels are connected with cross tunnels into pairs, as shown in the sketch. Connected to each of these eight pairs of branch tunnels is a 24,000 cfm centrifugal blower discharging to the power house and in the power house there are eight 20,000 cfm propeller type exhaust fans. It will be seen that the input capacity exceeds the outgo capacity, making it a pressure system.

The object in passing the air through the penstock tunnels first is to reduce the temperature 20 F. The mean temperature of the penstock is about 55 F and the tunnel walls are fairly constant at 80 F, giving a mean tunnel temperature of about 67.5 F which is approximately the mean outdoor air temperature (which varies the year around between limits of about 120 and 20 F). Air taken in at 120 F is reduced to 100 F simply by passing through the tunnel system and conversely, in the winter, air at 20 F is warmed to about 40 F by the tunnels.

Further warming of the air in the power house from 40 F is accomplished by electric space heaters, but 100 F air is considered entirely satisfactory for summer conditions in the power house, except for the offices and working balconies.

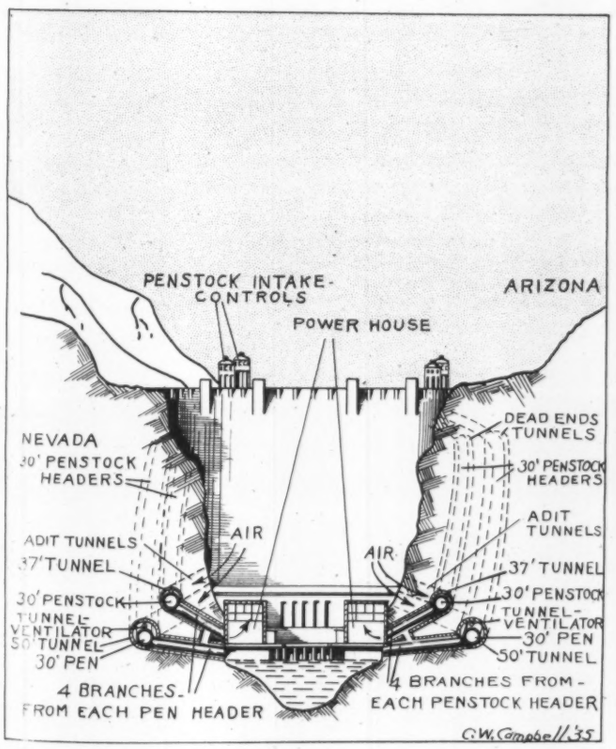
The offices, control rooms, motor-generator rooms and pipe shop are supplied with air by fans that pull air from the power house main enclosure through surface coolers through which penstock water at 60 F is circulated. The highest air temperature coming through these surface coolers is then, 100 F, and they each have a capacity of 20,000 cfm; the specifications call for a reduction in temperature of the air to a point where it will absorb 396,000 Btu per hr without exceeding 95 F at the fans. The coolers are made of copper finned tubing and a complete change of air is provided for every 10 minutes.

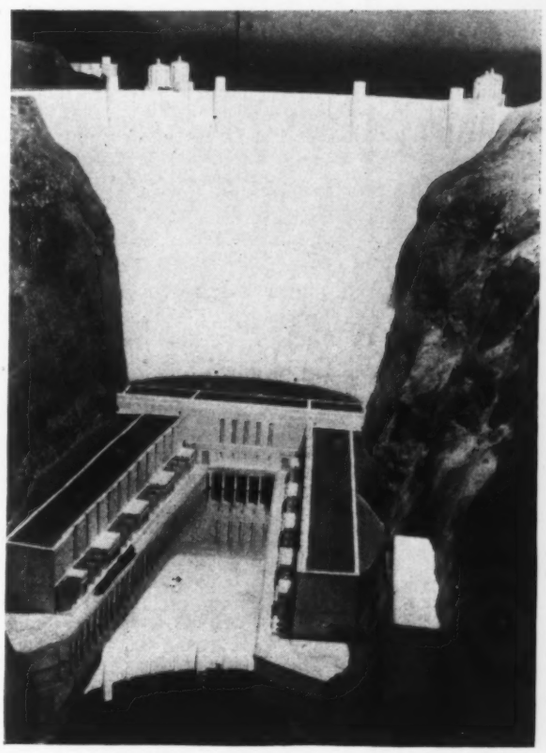
No attempt is made to supply artificial humidity. In the wash rooms, a complete change of air is provided every five minutes and in the power house at large, every hour.

In the winter, electrically heated water at 100 F circulated through the same surface heat exchangers warms the air taken through them from the power house main rooms at 40 F. Arrangements are provided whereby air may be taken in for the offices and work rooms directly from the outside when desired, instead of from the power house, by opening windows and closing up the surface heat exchangers.

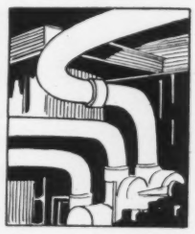
The author wishes to acknowledge the courtesy of the U. S. Reclamation Bureau personnel in making this information available.

The author's rough sketch shows a general cross-section of the Boulder dam power plant. Ventilating air is taken in through the adit tunnels, passes through the penstock tunnels and is discharged into the power house, being cooled or warmed 20 F during its passage through the tunnels. It is then drawn from the power house through surface coolers or heaters into the offices, control rooms and work shops. The dotted lines indicating the course of the penstocks and tunnels to the head water are merely diagrammatic. . . . The scale model shown was built by the U. S. Reclamation Bureau in their Denver laboratory. Length of each wing measured from the dam base to the end is 740 ft and the whole structure is about equal to a 12 story building five city blocks long.





Power Piping Requirements



By **J. Roy Tanner***

THERE was a time not long ago when power piping was divided into two broad classes. Small units and almost all industrial piping were designed to handle pressures usually under 250 lb per sq in. and steam in the saturated condition. Higher pressures with moderate superheat prevailed only in the larger central stations. Within the last fifteen years pressures and temperatures have been constantly stepped up as the pioneering of one installation became the standard practice of those following and the march of progress still continues.

The superposed plant has again broadened the field of high pressures and temperatures so that now, in addition to the central station demand for the highest grade of piping, every important industry is a potential user and the number of engineers who must be prepared to design and operate such equipment has vastly increased.

System design developments have been in the direction of increased reliability and consequently have greatly reduced the danger of shut down from failures in piping systems. The result has been the omission from many modern layouts of the elaborate cross-connections and valving arrangements mandatory before, with consequent reduction in piping cost per power unit.

The advance in the art of welding has also had a marked influence on layout and installation. Ponderous steel castings for complicated branch manifolds have given way to carefully designed welded construction and welding fittings and valves have made possible even an all welded, all field fabricated job where conditions warrant. The average layout for best economy is usually a compromise between the 100 per cent welded job and the job with no welding. To arrive at the best combination, comparative studies should be made taking all costs into account. Such studies will amply compensate for the time and effort spent.

The making of comparative estimates is not a matter of great difficulty. Fortunately the fabrication industry has progressed commercially as well as mechanically and there now exist, obtainable on request from any qualified fabricator, standard price lists covering all items encountered in the usual layout, such as bends of all types, lap joints of any description, line welds, welded branches, etc. These lists with the current discounts applied will produce a very close estimated cost for a shop fabricated job which can be compared with one for field work. In arriving at the latter figure for comparison a word of caution is in order. Code welding, that is welding the technique of which assures a 100 per cent joint, is of comparatively recent development. Its success depends upon deposition in a number of small beads and the process cannot be hurried. Much of the older published data on the time required for making welds was compiled with speed as the governing factor and cannot be depended upon for code welding.

Standardization: The *American Standards Association* is the fountain head of national engineering standardization. The work of three *A. S. A.* sectional committees has had profound influence on power piping, namely B16 on pipe flanges and fittings, B31 on a code for pressure piping and B36 on wrought iron and steel pipe. Sectional committee B31 began its work in 1926 and last year the Tentative American Standard Code for Pressure Piping was issued in printed form. This committee was a large group of exceptional ability which did a big job exceedingly well. (See articles by Sabin Crocker in HEATING, PIPING AND AIR CONDITIONING, January-August, 1934, for a complete explanation of this code.—ED.)

In a code of this wide scope, it is inevitable that some criticism should be evoked. Probably there are omissions even from a safety point of view and also probably there are prohibitions of practices which have caused no failures of record. Most of the adverse opinions I have heard are to the effect that the requirements are not strict enough, but these views are probably due to a misconception of the purpose of the code. It must be remembered that this work is a safety code first, last and all the time, and that much that we as engineers would not want to use in our plants, taking into consideration service plus reliability, cannot be condemned as unsafe. For instance, 125 lb fittings are allowable for steam at 125 lb and 450 F or standard weight pipe in some small sizes may be used up to 600 lb. Good practice often requires material of higher grade than safety alone would permit.

An important and necessary addition to the code is some safety provision for compensating for the loss of strength where pipe walls are cut for attachment of branches. A sub-committee is at work on this at the present time. It has been established by actual destruction tests that a welded branch up to 50 per cent of the run diameter does not weaken the wall, probably due to the effect of the branch as a collar and to the added metal in the fillets. The *A. S. M. E.* boiler code and the *A. P. I.-A. S. M. E.* unfired pressure vessel code both have rules for calculating reinforcement but it is by no means certain that these rules hold good for the relatively different proportions of run and branch encountered in pipe work.

Also it would be much better to establish definite dimensions for each case rather than to govern by means of a more or less complicated formula. A comprehensive and elaborate program of destruction tests of full size samples has been planned and will be carried out cooperatively by fabricators, pipe mills, and inspection engineers to establish definitely reinforcement proportions which will entirely compensate for the cut away wall. Some idea of the scope of this work can be gained when it is realized that the tests embrace enough samples to thoroughly survey each pressure range up to the largest sizes obtainable.

*Pittsburgh Valve, Foundry & Construction Co., Pittsburgh, Pa. Abstracted from an extensive paper presented at the Midwest Power Engineering Conference, Chicago, Ill.

Heating · Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

JULY 1936

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Application Factors Which Govern the Selection of Refrigerating Equipment for Air Conditioning Service

By J. R. Hertzler* *(MEMBER)*. York, Pa.

COMPRESSION type refrigeration equipment using dichlorodifluoromethane as the refrigerant is doubtless used in at least 90 per cent of all air conditioning installations made at the present time, for capacities up to 100 tons. There are no standards available which define the flexibility of operation required, the amount of condensing surface economically justified, the type of condenser and means of final heat rejection from the air conditioning system and the method of cooling and dehumidifying the air to be supplied to the conditioned space. It will be the purpose of this discussion to analyze the actual conditions of operation which a refrigeration system should be designed to meet when installed for human comfort air conditioning and to establish a basis for the proper balancing and selection of compressor, condenser and evaporator for usual conditions of operation encountered.

Refrigeration Load in Human Comfort Applications

Previous papers have described the cooling load factor in air conditioning, in which the refrigeration load was divided into its component parts:

1. Heat transmission through walls
2. Sun effect
3. Outside air load
4. People
5. Electrical load.

The first three factors; transmission, sun effect and outside air, will vary from zero to maximum, during the summer cooling season of four months. The internal loads, consisting of lights, people and any other source of internal heat such as fan motors, cooking equipment, escalator motors, or other equipment generating heat or moisture, are likewise variable but do not vary to the degree stated for the loads dependent upon the outside conditions.

The A.S.H.V.E. GUIDE 1936, Chapter 3, Table 2, specifies indoor temperatures recommended in summer, for various outdoor temperatures which conditions are applicable for exposure of less than 3 hours. This gives a graduated scale of indoor temperatures to be produced for varying outdoor temperatures, arranged so that the dew point is held constant in the conditioned space, during summer operation, for dry bulbs ranging from 80 F with 95 F outside, down to 72 F inside for a 70 F outdoor temperature. It has been the experience of operators of human comfort air conditioning installations, where the systems have been installed for their promotional value, to produce a differential in temperature between the outdoor and the indoor temperature. Therefore any analysis of load factor which is based on a constant indoor temperature, such as the design temperature for the maximum outdoor dry-bulb, is not a true consideration of the load factor in accordance with recommendations of the Society and is further not a true load factor in accordance with the usual method of operation adopted by owners of air conditioning plants. From a record of hourly outdoor dry-bulb temperatures, tabulated in a previous paper[1] the transmission load for a department store in New York City would be 486 hours out of a possible 1220 hours operation, provided the indoor dry-bulb design condition, in this case 80 F, were maintained throughout the summer operating period. On the basis of information obtained in this previous paper[1], the load factor for transmission would be 12.9 per cent, if the indoor dry-bulb temperature were maintained constant at the design condition of 80 F throughout the summer. In accordance with THE GUIDE recommendations, the indoor wet-bulb temperature would also be dropped as the dry-bulb is reduced for lower outdoor temperatures. Any consideration of load factor which is made with the assumption that the indoor wet-bulb temperature will be constant throughout the summer operating season, is not in accordance with actual practice. Similarly, if the indoor wet-bulb temperature were maintained constant at 67 F, which is the indoor design wet-bulb temperature used for a maximum outdoor wet-bulb temperature of 75 F, the refrigeration system would need to operate only 548 hours out of a total of 1220 hours summer operation of the equipment, and the load factor due to heat from outside air would be 25.3 per cent.

In the following examples, it is assumed that air will be supplied to the conditioned spaces at a constant dew point temperature of approximately 54 F to produce an indoor dew point temperature of 57 F, in accordance with Table 2 of Chapter 3 of THE GUIDE. On this basis, refrigeration will be required to cool the outside air, whenever the outdoor wet-bulb temperature is above 54 F. It is further assumed that only outside air would be passed through the dehumidifier when the outdoor wet-bulb temperature is lower than the indoor wet-bulb temperature. From an analysis of United States Weather Bureau records of New York City for the years

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[1]What Is the Cooling Load Factor in Air Conditioning? by John Everetts, Jr., A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.

1925 to 1931, the accompanying Table 1 indicates the number of hours from 8 a. m. to 5 p. m. daily, when wet-bulb temperatures between the limits tabulated occur. The average refrigeration load for the outside air and the ton hours of refrigeration per season for each of the groups of outside wet-bulb temperatures are also tabulated.

Table 1—Load Variations in Cooling Outside Air to 54 F Wet-Bulb

Outside Wet-Bulb Temp.	Hours	Ave. Load in Tons	Ton Hours
79F-75F	105	100.0	10500
74 -70	275	74.5	20500
69 -65	330	50.6	16700
64 -60	275	29.1	8000
59 -55	160	10.4	1665
Total .....1145 hours			57365

Maximum Load at 77 F wet-bulb = 100 tons.

$$\text{Load Factor Outside Air} = \frac{57365 \text{ ton hours}}{1145 \text{ hours} \times 100 \text{ tons}} = 50 \text{ per cent.}$$

Dividing the operating season of the refrigeration equipment into five periods for outside wet-bulb temperatures as tabulated, that is, from 79 to 75 F, from 74 to 70 F, etc., the refrigeration load in ton hours per year required for cooling outside air would be 57,365 ton hours, out of a possible total of 114,500 ton hours for 1145 hours total operation, which results in a load factor of 50 per cent as indicated. This method of estimating load requirements due to outside air generally produces a result which is higher than the actual due to the fact that the final wet-bulb temperature calculated leaving the equipment is lower than the indoor wet-bulb temperature. However, the basis used is one which is believed to more nearly approach the actual than any form of analysis based on maintaining the inside wet-bulb temperature at the design condition for the assumed outdoor maximum. Another example will indicate a comparison using the actual indoor wet-bulb temperature rather than the apparatus temperature as the base for determining load factor due to heat added by outside air.

The method of obtaining the overall load factor of an office building or department store internal load, which constitutes principally the electrical and people load, assuming population densities and lighting loads on a unit of square feet of floor area of the conditioned space, is shown in Fig. 1. Knowing the electric lighting load plus internal electric load in watts per square foot, on which the load factor is 100 per cent and further knowing the square feet of floor area per person in the conditioned space, as well as an approximate load factor on the average number of people as compared to the maximum number of people for a given space, the overall load factor on light and people or the overall internal load factor may be determined directly from the curve. For example, with 2 watts of electric load per square foot, 40 sq ft of floor space per person and 25 per cent load factor for people, the overall internal load factor obtained directly from the curve would be 55 per cent. The curve indicates the load factors of three different factors for people loads of 25, 50 and 75 per cent. For other load factors on population, it is necessary to interpolate between the given load factors and the given electrical load curve.

Knowing the division of internal load, the load on a given job due to outside air and heat transmission, and the load factor on the internal load, it is possible from Fig. 2 to obtain directly the overall load factor for a particular installation. Thus with a 55 per cent load factor for an internal load, as determined from the previous example and a division of 40 per cent outside air and transmission load and 60 per cent internal load, the overall load factor for the plant under consideration would be 53 per cent. This value is obtained by reading vertically upward on the 40 per cent outside air and transmission load line to the 55 per cent load factor on internal load line, intersecting at 53 per cent overall load factor. In the department store example analyzed in a previous paper[1] the internal load on the department store was 80.5 per cent of the total and the outside air, transmission and sun effect were 19.5 per cent of the total.

It is proposed that the assumption be made that the transmission load and sun effect load vary approximately in the same proportion that the outside air load as measured by the wet-bulb temperature, assuming a constant dew point of 54 F leaving the air dehumidifier. For the condition of operation, as previously explained, the outside air load, transmission load and sun effect would have an overall seasonal load factor of approximately 50 per cent for most human comfort applications which may be extended to include eastern and central United States, excluding only the southern and far western portions of this country. On the basis of the above assumption, the load factor on human comfort air conditioning jobs approaches 50 per cent as the proportion of internal load to total load decreases and falls between 50 per cent and the load factor on the internal load, for conditions of operation where the load factor on the internal load of a given installation is in excess of 50 per cent.

This analysis and the comparison with supplementary curves may be used only for buildings cooled during the period between 8 a. m. and 5 p. m. It is necessary to separate loads individually and analyze the individual condi-

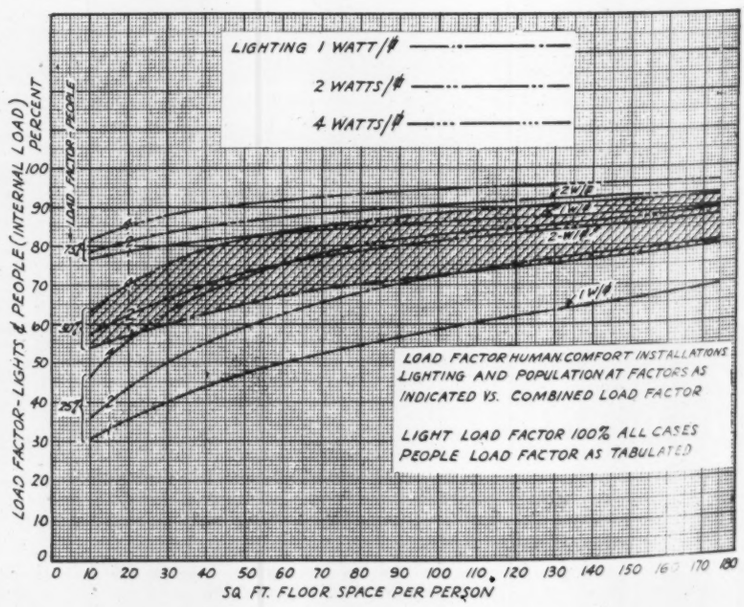

LIGHTING 1 WATT/ft
2 WATTS/ft
4 WATTS/ft
LOAD FACTOR-LIGHTS & PEOPLE (INTERNAL LOAD) PERCENT
LOAD FACTOR-PEOPLE PERCENT
LOAD FACTOR HUMAN COMFORT INSTALLATIONS LIGHTING AND POPULATION AT FACTORS AS INDICATED VS. COMBINED LOAD FACTOR
LIGHT LOAD FACTOR 100% ALL CASES PEOPLE LOAD FACTOR AS TABULATED
SQ. FT. FLOOR SPACE PER PERSON

Fig. 1—Internal load factor curves for various floor spaces, occupancies, and light intensities

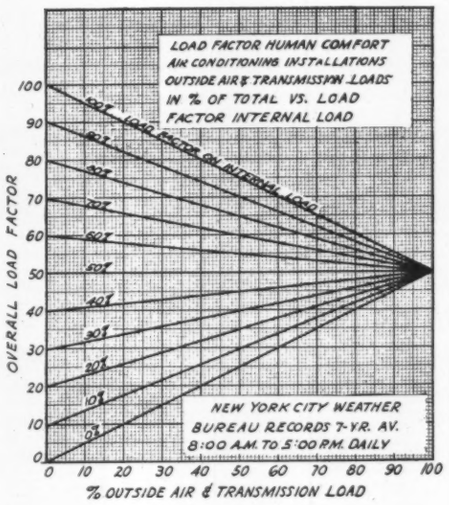

LOAD FACTOR HUMAN COMFORT AIR CONDITIONING INSTALLATIONS OUTSIDE AIR & TRANSMISSION LOADS IN % OF TOTAL VS. LOAD FACTOR INTERNAL LOAD
100% LOAD FACTOR ON INTERNAL LOADS
80%
60%
40%
20%
0%
NEW YORK CITY WEATHER BUREAU RECORDS 7-YR. AV. 8:00 A.M. TO 5:00 P.M. DAILY
OVERALL LOAD FACTOR
% OUTSIDE AIR & TRANSMISSION LOAD
0 10 20 30 40 50 60 70 80 90 100

Fig. 2—Overall load factor curves for various outside air, transmission and internal load factors

tions for theaters or restaurants, when the operating hours are different from 8 a. m. to 5 p. m. daily.

Types of Refrigerating Equipment

Condensing Units

In the small sizes practically all manufacturers offer a line of standardized self-contained automatic dichlorodifluoromethane condensing units, including an electrically driven compressor with a condenser of the shell and coil, double pipe, or shell and tube type, mounted in or on the compressor subbase. These machines are driven by electric motors at speeds in accordance with load requirements and the condensing surface utilized in the refrigerant condenser is proportioned so as to be applicable to a wide range of operating conditions. Operation of condensing units at evaporator temperatures used in human comfort air conditioning applications impose the greatest load on the refrigerant condensers. The refrigeration effect of the condensing unit is increased for increases in evaporator temperature, thus allowing a smaller amount of condensing surface per ton of refrigeration effect for air conditioning, than would be the case should the same condensing unit be applied to cool a low temperature meat box maintained, for example, at 38 F.

Large Compressors

Above 40 hp, dichlorodifluoromethane compressors may be furnished with various types of drives, including electric motor, either direct connected to large synchronous motor driven compressors, or v-belt drive. It is likewise possible to operate large compressors by direct connection to diesel engines, gas engines or steam engines of the reciprocating type. Where operating economy dictates the use of low pressure or high pressure steam turbines, this type of drive can be applied to a reciprocating type compressor, by means of a combination speed reduction gear and belt drive assembly.

Because of the fact that refrigeration equipment is not uniformly rated at definite conditions as to evaporator and condensing temperature, it is difficult to rate a machine for a specific capacity, unless the actual operating conditions at which this capacity is obtainable are stated. It is desirable that the *American Society of Refrigerating Engineers* establish a method of testing and a method of rating dichlorodifluoromethane compressors at given standard air conditioning evaporator and condenser temperatures, so that some of the confusion now prevalent may be eliminated. At the present time a proposed Standard Method of Rating and Testing Mechanical Condensing Units has been formulated for small units, but as yet no published ratings are available. The variation in capacity in tons of refrigeration is indicated in Fig. 3 for a given dichlorodifluoromethane machine, operated at a fixed speed and handling a fixed volume of gas. For condensing temperatures of 70 to 120 F and for a 40 F evaporator temperature, the capacity of this compressor varies from 125 to 90 per cent

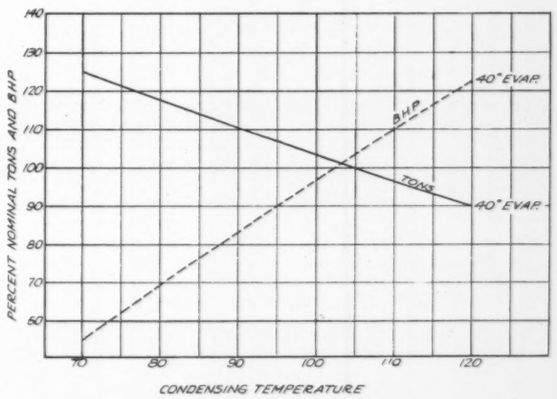


Fig. 3—Capacity variations of compressors operating at a fixed speed and constant gas volume

nominal tons. Throughout the same range of condensing temperature operation the brake horsepower varies from 55 to 123 per cent bhp nominal. In the lower portion of Fig. 4 tons of refrigeration and brake horsepower at a constant condenser temperature of 100 F for varying evaporator temperatures from 20 to 50 F are shown. It is therefore apparent that the compressor operated by the same motor at 20 F would have a capacity of 66 per cent nominal tons of refrigeration when requiring 96 per cent nominal brake horsepower and would have a capacity of 127.5 per cent nominal tons of refrigeration and require only 93 per cent nominal brake horsepower at 50 F evaporator temperature. The absolute necessity of fixing the operating conditions for a given compressor is indicated by reference to these curves. The upper portion of Fig. 4 shows the cost of power in cents per ton hour for a wide range of evaporator and condensing temperature conditions at which the compressor is operated. These power costs are tabulated for a unit price of 1, 2 or 3 cents power cost per kilowatt hour. A further application of these curves will be illustrated in a later example.

Condensers

In refrigeration systems, other than the unit type, condensers of various types, sizes and capacities are frequently used with large compressors as previously described. In the past large installations of condensers of the shell and tube type have been used where a supply

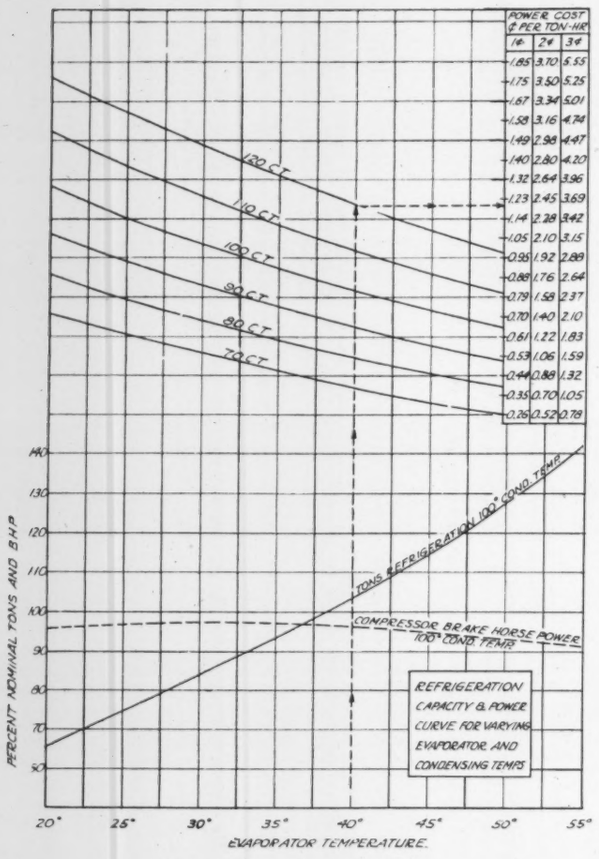


Fig. 4—Refrigeration capacity and power curves for varying evaporator and condensing temperatures

of condensing water was available, either from city mains, deep wells or cooling tower systems. These condensers were generally designed so that they could be readily retubed in the field and were therefore installed using bare pipe surface of steel or non-ferrous materials. While it is true that the condensing temperature obtainable with various types of condensers is not always directly in proportion to the amount of condensing surface installed in a given condenser, Fig. 5 indicates an approximate operating condition obtainable under normal conditions of operation for varying quantities of condensing surface installed. The difference between the temperature of the water leaving the condenser and the refrigerant condensing temperature obtained for varying quantities of condensing surface is shown in Fig. 5. From this curve it is apparent that the proper proportion of condensing surface to be used in connection with a large refrigeration system is an economical consideration. An increased amount of condensing surface will provide a reduced operating cost for a dichlorodifluoromethane refrigeration system and will involve an increased first cost to the consumer. The limit to which condensing surface should be added for improving operating economy is thus dependent upon the power saving which would be obtainable with a larger condenser. It is likewise apparent that the condensing surface furnished for a given compressor would vary in accordance with the cost of power on individual installations and would vary for cooling tower installations as compared to city water installations.

A basis for determining the economical limitations of design and cost are shown in Fig. 6 which indicates the difference between the final condensing water temperature leaving a cooling tower and the outside wet-bulb temperature entering the cooling tower, for various conditions of load. This curve is based upon the assumption that at no load, the water would leave the cooling tower 2 F above the entering wet-bulb temperature, whereas for full load the difference between the leaving water and entering wet-bulb temperature would be 8 F. Naturally these conditions vary in accordance with the tower design but they are approximate average values which may be used for this basis of comparison. For conditions of less than full load, the difference between leaving water and entering wet-bulb temperature varies in direct proportion to the load on the tower. The analysis of an individual problem of condenser selection for a cooling tower installation of 100 tons capacity, operated from 8 a. m. to 5 p. m. in the eastern territory, with a load factor of 60.5 per cent overall is graphically shown in the curves of

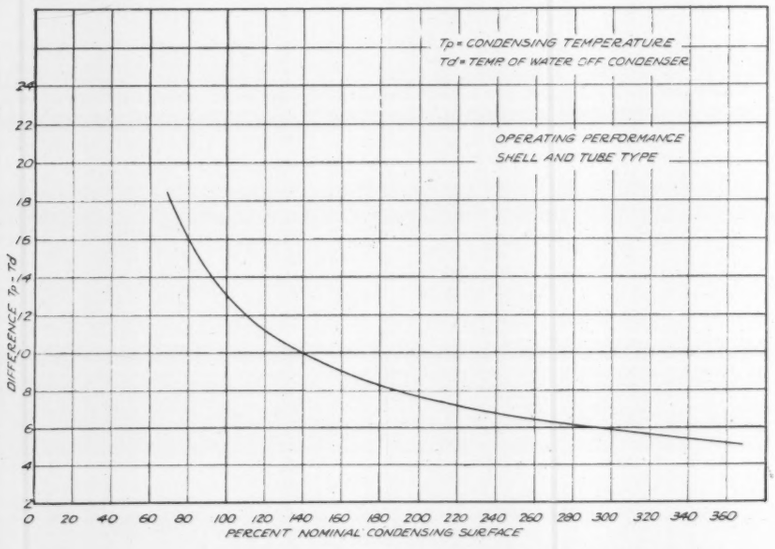

Tp = CONDENSING TEMPERATURE
Td = TEMP OF WATER OFF CONDENSER
OPERATING PERFORMANCE
SHELL AND TUBE TYPE
DIFFERENCE Tp - Td
PERCENT NOMINAL CONDENSING SURFACE

Fig. 5—Operating performance curve for shell and tube type of condenser

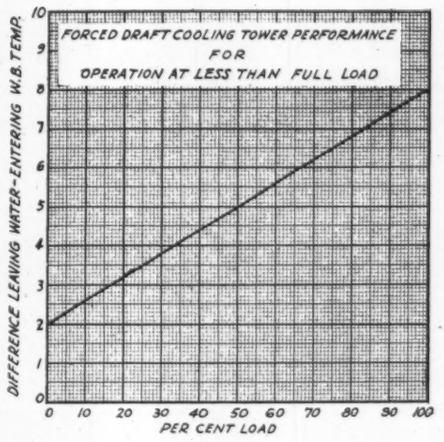

FORCED DRAFT COOLING TOWER PERFORMANCE
FOR
OPERATION AT LESS THAN FULL LOAD
DIFFERENCE LEAVING WATER-ENTERING W.B. TEMP.
PER CENT LOAD
0 10 20 30 40 50 60 70 80 90 100
0 1 2 3 4 5 6 7 8 9 10

Fig. 6—Forced draft cooling tower performance curve for varying load capacities

Fig. 7. This installation at maximum load required 100 tons of refrigeration, of which 47.5 per cent of the total is due to outside air and transmission, and 52.5 per cent of the total is internal load on which there is an overall internal load factor of 70 per cent. In this example a nominal condenser rating of 10 sq ft of condensing surface per ton was used. From an analysis of these curves it will be noted that for power at 1 cent per kwhr, 8 sq ft of condensing surface per ton is most economical. For power at 2 cents per kwhr, 9 sq ft of condensing surface per ton provides the lowest annual fixed and operating cost for the condenser. For a power cost of 3 cents per kwhr, 10 sq ft of condensing surface per ton is most economical. For power at 4 cents per kwhr, condensing surface in the amount of 10 sq ft per ton may be justified on the basis of this analysis of fixed and operating costs on the condenser. The proportion of fixed charge to operating cost may be noted by comparing the fixed charge curve shown at the bottom in dotted line with the total charge as indicated by full line on the cooling tower system.

For purpose of comparison, the total annual fixed and operating cost of a condensing system when utilizing city water with a final temperature leaving the condenser of 93 F throughout the year is shown for a 1 cent power cost. It will be noted that the power required by the refrigerating compressors is 30 per cent greater utilizing city water maintained at a constant discharge temperature of 93 F, than would be the case with a cooling tower system of the forced draft type utilizing outside air for cooling the condensing water in the system. This power saving for the compressor would be offset by the water pumping cost required for circulating the condenser water through the condenser and the cooling tower and would further be offset by the fan horsepower utilized by the fans for a cooling tower of the forced draft type. Since the condensing water cost was not included in this comparison, the economy of installing a cooling tower is not established by this analysis. It is generally true, however, that the power required for operating the refrigeration compressor in an air conditioning system is less for a cooling tower installation than would be the case with a city water system, in which the amount of city water used is controlled from the discharge temperature of the city water leaving the condenser.

The steps necessary in making a complete condenser cost analysis are indicated in Table 2.

Table 2—Cost Analysis of Cooling Tower Application

Wet-Bulb Temp. Range	Tons Refrig.	Diff. Water Leav. Tower and Wet-Bulb Temp. (Fig. 6)	Temp. Water Leav. Cool. Tower	Cool. Tower Water Range for 300 gpm.	Tp-Ta (Fig. 5) 80 Per Cent Nom. Sq Ft	Cond. Temp.	Evap. Temp. 40 F at 100 Tons	Cents/Ton Hour 1 Cent /Kwhr (Fig. 4)	Ton Hours	Operating Costs
79-75	84.25	7.1	84.10	8.4	13.7	106.2	41.6	0.88	8850	$ 77.80
74-70	72.15	6.5	78.30	7.2	11.9	97.4	42.8	0.73	19820	144.80
69-65	60.75	5.65	72.65	6.07	10.4	89.1	43.9	0.58	20000	116.00
64-60	50.55	5.05	67.05	5.06	9.1	81.2	44.9	0.46	13900	64.00
59-55	41.75	4.5	61.50	4.2	7.9	73.6	45.8	0.35	6680	23.40
										$426.00

Wet-Bulb Temp. Range	Water Leav. Cond.	Tp-Ta 90 Per Cent Nom. Sq Ft	Cond. Temp.	Cents/Ton Hour	Operating Costs	Tp-Ta 100 Per Cent Nom. Sq Ft	Cond. Temp.	Cents/Ton Hour	Operating Costs
79-75	92.5	12.25	104.8	0.87	$ 77.00	11.3	103.8	0.84	$ 74.30
74-70	85.5	10.8	96.3	0.70	139.00	10.1	95.6	0.68	135.00
69-65	78.7	9.5	88.2	0.57	114.00	8.8	87.5	0.56	112.00
64-60	72.1	8.3	80.4	0.44	61.20	7.7	79.8	0.44	61.20
59-55	65.7	7.3	73.0	0.34	22.70	6.7	72.4	0.34	22.70
					$413.90				$405.20

Wet-Bulb Temp. Range	Water Leav. Cond.	Tp-Ta 120 Per Cent Nom. Sq Ft	Cond. Temp.	Cents/Ton Hour	Operating Costs	Tp-Ta 150 Per Cent Nom. Sq Ft	Cond. Temp.	Cents/Ton Hour	Operating Costs
79-75	92.5	9.8	102.3	0.83	$ 73.50	8.3	100.8	0.80	$ 70.80
74-70	85.5	8.8	94.3	0.67	133.00	7.4	92.9	0.65	129.00
69-65	78.7	7.7	86.4	0.55	110.00	6.6	85.3	0.53	106.00
64-60	72.1	6.8	78.9	0.43	59.80	5.9	78.0	0.42	58.40
59-55	65.7	6.0	71.7	0.33	22.10	5.1	70.8	0.32	21.40
					$398.40				$385.60

For the five groups of wet-bulb temperatures it was necessary to establish the refrigeration load at the average wet-bulb temperatures for each group. The difference between the water leaving the tower and the wet-bulb temperature is shown in Fig. 8, which establishes the leaving water temperature of the tower. The difference between condensing temperature and temperature of the water leaving condenser may be obtained from Fig. 5. The cooling tower water range is in proportion to the refrigeration tonnage for the particular load condition with the maximum load imposing a range of 10 F on the quantity of condensing water circulated. This condition would be on the basis of 3 gpm per ton, which would increase the temperature through a 10 F range on ordinary air conditioning applications. By adding the difference in temperature values obtained from Fig. 5 plus the cooling tower water range to the water temperature leaving the cooling tower, the refrigerant condensing temperature is obtained. The evaporator temperature in each case is in accordance with the refrigeration load in the water cooler. For a full load of 100 tons, the temperature difference between the average chilled water, which is assumed to be 50 F, for producing a 54 F leaving air temperature at the dehumidifier is 10 F. That is, the amount of surface in the water cooler is proportioned so that with an evaporator temperature of 40 F and an average chilled water temperature of 50 F the cooler will have the required capacity of 100 tons. Then for each

condition of load as tabulated, the temperature difference would be in proportion to the load on the cooler. The power cost in cents per ton hour is selected for the exact evaporator and the condenser temperatures obtained for each case in accordance with values given on Fig. 4 for power at 1 cent per kwhr. The ton hours of refrigeration load is the product of the tons refrigeration load for the various wet-bulb temperatures under consideration and the number of hours at which these wet-bulb temperatures occur per year. The operating cost for each wet-bulb condition is the product of the ton hours and the cost of refrigeration power in cents per ton hour, as tabulated.

Combined Cooling Tower Condensers

Where a forced draft tower is applicable or where it is desired to locate a cooling tower indoors, it is possible to build the refrigerant condensing coils in the cooling tower

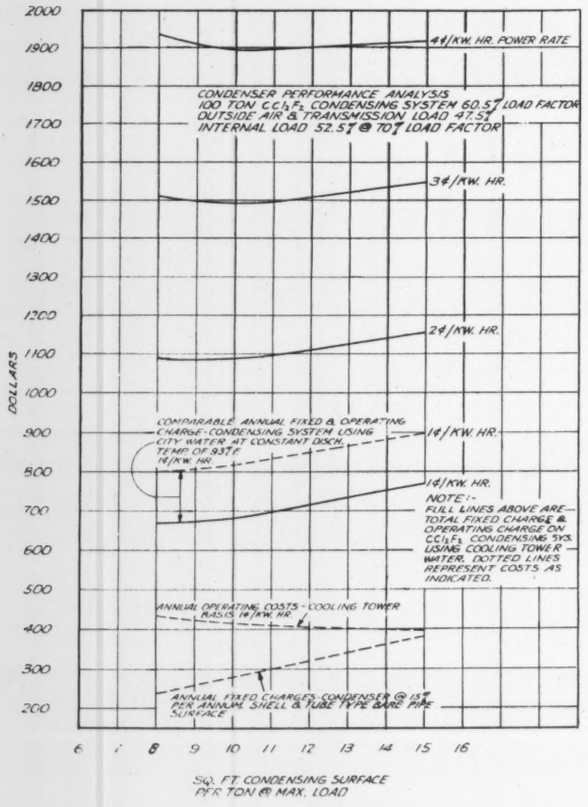


Fig. 7—Economic condenser selection curves for different surface areas and power costs

spray chamber which reduces the usual water pumping cost required in an ordinary forced draft cooling tower water system. The fan horsepower would be comparable to a forced draft cooling tower of the usual design and condensing temperatures obtainable are dependent upon the amount of condensing surface installed in the combined cooling tower condenser per ton of refrigeration, and the final wet-bulb temperature of the air leaving the combined cooling tower condenser.

There are many variable factors affecting the design of a combined cooling tower condenser and a definite basis for rating this type apparatus is of even greater importance than is the case in other air conditioning equipment.

The capacity of a given cooling tower condenser varies with changes in each of the following operating conditions:

1. Evaporator temperature.
2. Condensing temperature required.
3. Entering air wet-bulb temperature.
4. Air volume supplied and final wet-bulb temperature of leaving air.

For example, a unit cooling tower condenser may be rated for a given capacity with an evaporator temperature of 30 F, a condensing temperature of 110 F, an entering air wet-bulb temperature of 75 F and a fixed quantity of air supplied for the final means of heat rejection from an air conditioned space.

Capacity changes due to operation at other than rated evaporator temperatures are small, with a 7.5 per cent increase in capacity for a 30 F rise in evaporator temperature to 60 F, and an 8.5 per cent decrease in capacity for a 30 F reduction in evaporator temperature to 0 F.

Rated capacities are related to the condensing temperatures to be produced in the condenser of a dichlorodifluoromethane refrigeration system. If an outside wet-bulb temperature of 75 F and a fixed quantity of air for a cooling tower condenser were required to produce a condensing temperature of 90 F instead of 110 F rated, its capacity would be 38.4 per cent of the nominal capacity at 110 F condensing temperature. Were the condensing temperature permitted to rise to 120 F instead of 110 F the nominal rated capacity of the apparatus would be increased 35.5 per cent all other factors remaining constant.

The variation in capacity of a combined cooling tower condenser with changes in the wet-bulb temperature of the entering air is shown in Fig. 8. If the wet-bulb temperature at which the equipment is rated were dropped from 75 to 65 F the capacity would be increased 17.5 per cent all other variables remaining constant and if raised from 75 to 85 F the capacity would be decreased 22 per cent.

Other factors governing the design of this type of equipment have a definite bearing on capacity such as the type of water distribution whether sprayed or of the Baudelot type, air velocities over condensing surface, depth of condensing surface in direction of air flow or actual type of condensing surface whether finned or plain pipe.

The ease of cleaning and maintenance with this piece of equipment is exceedingly important inasmuch as the source of air supply may be contaminated to a considerable degree.

It is of greater importance when making operating power comparisons to consider the condensing temperature rating of the cooling tower condenser rather than the actual fan and motor pump horsepower requirements. In most cases the power requirements of the refrigeration compressor will be ten or more times that of the cooling tower condenser so that economies of operation may more readily be obtained by reducing condensing temperature than would be possible by refinements of fan and pumping arrangements. Due to the relatively greater importance of the heat of the liquid of dichlorodifluoromethane a reduction in condensing temperature will likewise cause an increased capacity of the refrigeration compressor in addition to a decreased power input to the compressor motor. Therefore a reduction in condensing temperature is of greater importance than subcooling of

the refrigerant because of reduced power requirements as well as increased capacity as the compressor displacement remains constant.

Evaporators

Evaporators may be furnished for dichlorodifluoromethane air conditioning applications of the direct expansion fin surface type. Fin surfaces are furnished in many styles and types, such as helical, flat, corrugated, and compound fins, which are usually bonded to the pipe coils by means of a tinning operation. In most cases the thermostatic type of expansion valve is used for fin surface direct expansion air conditioning applications.

It is likewise possible, in the application of air cooling with direct expansion, to locate the cooling coils in the spray chamber of an air dehumidifier. This method of design obtains all of the advantages of a spray type dehumidifier, without enlarging the cross section of the equipment, provided the evaporator coils are properly designed. Because of the high heat transfer obtained with a water spray on the exterior surface of the direct expansion evaporator piping, fins are of no particular advantage, with the result that bare pipe coils have an installation advantage in the spray type chamber of air dehumidifiers. The flooded evaporator method of operation is preferable to thermostatic expansion, when the coils are located in the water spray of a dehumidifier, because of the smaller temperature difference used for economical compressor selection and operation. One of the principal advantages of a dehumidifier system is that the air leaves the apparatus in a saturated state, so that the relative humidity in the conditioned space is more closely controlled. Furthermore a duct system designed for the same inside temperature and humidity may be smaller in area with an air dehumidifier of the spray type than with a surface coil type installation. In the intermediate seasons, that is, spring and fall operation of the air conditioning equipment, the sprays of the air dehumidifier can be used to obtain evaporative cooling on days when the operation of the mechanical refrigeration system would be required with a surface cooling installation.

The indirect method of refrigeration may similarly be applied to both surface cooling and air dehumidifier spray type coolers, with the water cooled by a separate refrigeration plant and circulated to the air cooling surface or the air dehumidifier. The majority of all human comfort air conditioning installations made to date in the large sizes have been designed with a water spray in preference to the surface cooling type of installation. However, surface cooling has become more popular in the past few years.

Application of Refrigeration Equipment

From the previous discussion it is evident that a thorough analysis of the load factor is necessary before the proper selection of refrigeration equipment may be made for an air conditioning installation.

The minimum capacity at which a plant will be required to operate will determine the minimum number of refrigeration compressors necessary or will dictate the number and types of capacity reducing devices which will be installed for a given compressor as selected. Most large dichlorodifluoromethane reciprocating compressors are equipped with hand or automatically operated capacity reducing devices which will permit economical operation at partial loads down to approximately 50 per cent of the maximum capacity. If the load analysis indicates that a further reduction in capacity will be necessary for certain periods of operation this can be applied to a single compressor installation by furnishing speed reduction on the compressor motor down to 50 per cent of maximum speed, thus reducing the capacity of the compressor to 25 per cent of maximum by the combination of capacity reducing devices on the machine itself and reduced speed operation. Adjustable speed control is especially desirable on an installation where the driving motors operate on direct current because of the economical operation obtainable by means of field control of a direct current adjustable speed motor. On alternating current installations multispeed motors are applicable up to 150 hp and where direct connected synchronous motors are used it is desirable to increase the number of compressor units to obtain the minimum refrigeration capacity and flexibility of operation required.

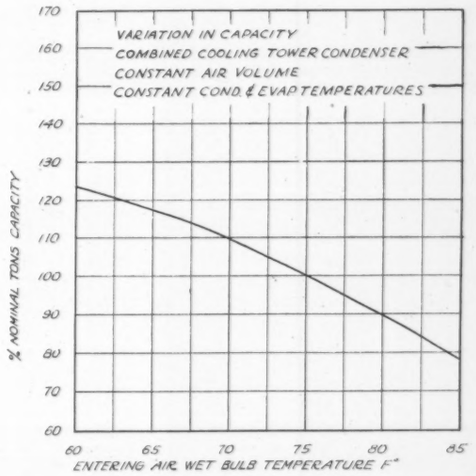


Fig. 8—Capacity variation curve for combined cooling tower condenser

Where refrigeration units are installed in multiple for a given installation it is generally advantageous to have the compressors installed so that all of the condensing surface and all of the evaporating surface can be utilized for all conditions of operation, which results in a direct operating economy as shown in Table 2.

It has been indicated that the kilowatt input per ton to an electrically driven dichlorodifluoromethane compression type refrigeration system is reduced for partial load operation provided the full condensing and evaporator surface is utilized at all loads. It was shown in a previous paper[2] that economy dictated the use of 9 sq ft of condensing surface per ton. When operated at ½ capacity, with the evaporator temperature held constant, the brake horsepower per ton of refrigeration was reduced 9 per cent. By raising the evaporator temperature 5 F at ½ load operation, with the condensing temperature held constant, the brake horsepower per ton would be decreased 9.5 per cent or combined, the saving by using the full condensing and evaporating surface for operation at ½ load would be 17.7 per cent in brake horsepower per ton. Since the compressor motor

[2]Operating Results of an Air Conditioning System Compared with Design Figures, by J. R. Hertzler, A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934.

efficiency was 4 per cent lower at ½ speed than at full speed the full saving of 17.7 per cent was not obtained but the net saving was sufficient to justify the installation of a single cooler and condenser to be used with a variable compressor capacity. Operating costs may be reduced as much as 25 per cent by utilizing this arrangement of equipment selection.

It is therefore desirable to parallel condensers or evaporators, or to install more than one compressor to operate on a given condenser and water cooler. This method of application may be considered a standard arrangement of design because of the resultant reduced operating costs when using dichlorodifluoromethane equipment. The operating economy of a refrigeration system at partial load is far more important than the economy at full load because the refrigeration system operates at less than full load more than 90 per cent of the time.

Conclusions

As a basis for the selection of dichlorodifluoromethane refrigeration equipment for air conditioning service the following standards may be established.

1. The load factor and the actual loads at which the equipment will be required to operate determines the number of compressor units or the capacity regulating devices of the individual compressor units required.
2. The fixed charges on a particular refrigeration equipment should be calculated at 15 per cent of the initial cost per annum.
3. The cost of the driving energy and the first cost of the prime mover of the compressor generally determines the type of drive selected.
4. The amount of condensing surface justifiable is variable and dependent upon operating costs or the balance of first and operating cost which results in the lowest annual expense to the owner.
5. Use of combined cooling tower condensers may be justified on the basis of reduced annual fixed and operating charges or by other causes such as inadequate or improper condensing water supply or limited sewer capacity.
6. Evaporators are generally selected to provide the desired accuracy of control, flexibility of operation, ease of installation and the lowest seasonal fixed and operating charges.
7. As practically all systems must be operated at less than full load the greater part of the time, a method should be provided to operate the compressor economically at reduced capacity. Where flexibility of operation is provided by multiple compressor units, the evaporating and condensing surface installed with such units should be available for all conditions of load to provide the greatest operating economy.

Survey of Engineers

The June issue of the *Monthly Labor Review*, published by the U. S. Department of Labor, contains the first summary of the statistics from the Survey of Engineers conducted by the Bureau of Labor Statistics in August and September 1935 under the supervision of the Committee on Engineering and Allied Technical Profession of the American Engineering Council in cooperation with more than 100 national, state and local engineering societies. More than 60,000 questionnaires were returned to the Bureau of Labor Statistics for classification and codification.

The article briefly summarizes the number of engineers in each branch of the profession, indicates the number employed in government and in private industry, and shows the interrelation between their educational training and positions. Reprints of this article will be available early in July.

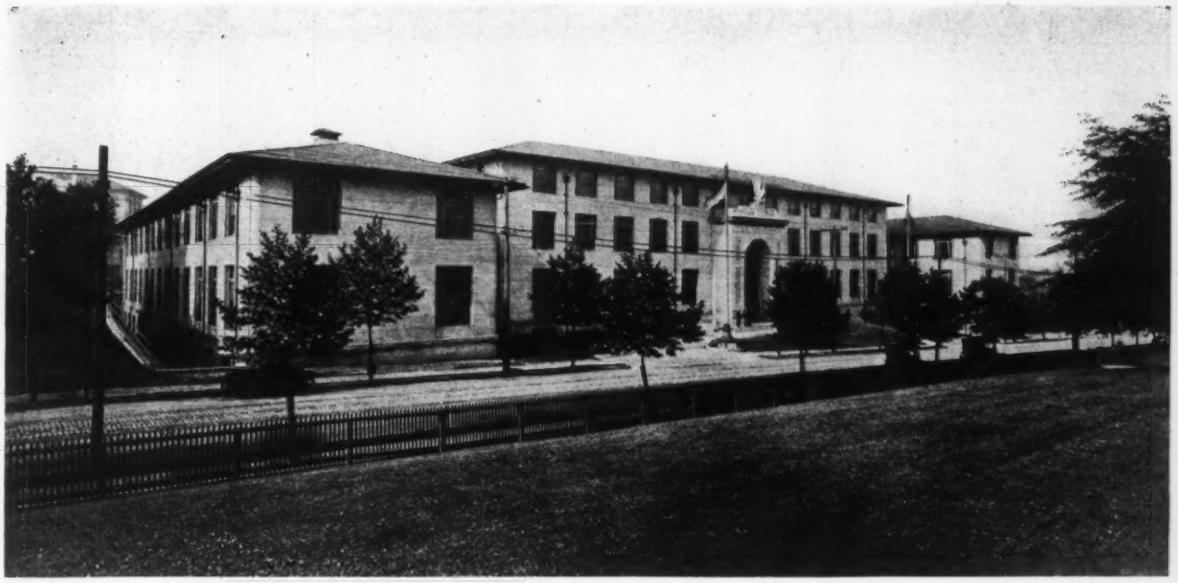
Census Bureau Reports Air Conditioning Sales

According to the report recently issued by Director William L. Austin of the Bureau of Census, Washington, D. C., the statistics on the value of orders booked for the first four months of this year indicate that the total air conditioning systems and equipment sales amounted to $11,667,129. The total of these orders recorded was from 98 manufacturers who represent a new series resulting from the expansion of the scope of inquiry. The total value of orders tabulated by these same manufacturers during March of this year amounted to $3,477,752 as compared with $2,611,241 for February, 1936, which is an increase of 33.2 per cent.

Approximately 175 manufacturers claim that they make air conditioning systems or equipment and accessories for complete installations, such as fans, coils, condensing units, evaporators, control equipment, air washers, filters, humidifiers, and heating auxiliary apparatus. In 1935 this same group of manufacturers had a total record of orders booked amounting to $48,244,800, which are divided into the following groups of applications:

	Per cent
Small Stores and Shops	21.9
Large Stores, Theatres and Public Buildings	19.9
Industrial Installations	19.7
Complete Residential All-Year Equipment	17.9
Railway Cars	10.7
Single Room or Office Units	9.9
	100.0

At the present time the Bureau of Census is segregating the monthly statistics for air conditioning systems and equipment into various accessory groups.



Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located

Heating Requirements of an Office Building as Affected by Weather Conditions

By F. C. Houghten* *(MEMBER)* and Carl Gutberlet** *(NON-MEMBER)*
Pittsburgh, Pa.

HEAT requirements of buildings have been the subject of intensive study by the Research Laboratory and cooperating institutions for several years. This investigation has included studies of the overall heat requirements of various groups of buildings calculated from the steam supplied by a central station, the heat requirements of individual buildings based on their steam requirements, and more intensive studies of the heat requirements of individual rooms in a building. This work has been carried on through 1935 under D. S. Boyden, then chairman, and more recently under O. W. Armspach, the present chairman of the Technical Advisory Committee on Heat Requirements of Buildings.

Included in this general investigation was an intensive study of the heat loss from eight rooms on three different floors and three different exposures of the Grant Building, Pittsburgh. The observations were made during the latter half of the 1933-34 heating season. A portion of the results were immediately analyzed and a report[1], dealing primarily with the relation between wind velocities as reported by the Weather Bureau and as observed near the building, was presented to the Society at the 1934 Semi-Annual Meeting. This present report is based on further analysis of the test observations and deals with the relation between the heat requirement of the several rooms and the weather conditions, including outside temperature, wind velocity and direction, and solar radiation. The effect of infiltration due to wind velocity, temperature difference and elevation in the building will be only briefly discussed in this paper. A more comprehensive analysis of this phase of the subject will be presented in a future report.

A complete description of the location, orientation and surroundings of the Grant Building, and the locations in the building of each of the rooms studied, as well as a detailed description of the arrangement of the test apparatus and methods used in collecting the data, are included in the earlier report[1], and will be only briefly reviewed here.

The Grant Building is a 40-story, modern office building of the set-back skyscraper type, built in 1926 and located in downtown Pittsburgh. The location in the building and the exposure of the eight rooms studied are given in Table 1. Each room was heated between approximately 8:00 a. m. and 10:00 p. m. by electrical

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[1]Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet. A.S.H.V.E. TRANSACTIONS, Vol. 40, 1934, p. 387
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1936, Buck Hill Falls, Pa.

resistance heating units placed below the steam heating units in the convector cabinets under the windows, so that the convected heat was supplied to the rooms through the same grille and with approximately the same air temperature, velocity and direction as occurred in the normal heating of the building with steam. The electrical input was indicated by integrating watt-hour meters, read at frequent intervals to an accuracy which gave the heat input to within plus or minus 17 Btu. Approximately two-thirds of the required electrical input was hand controlled and on continuously. The remaining required heat input was controlled by an *on* and *off* thermostat. Each room was at all times controlled to the same average temperature as the surrounding space in the building, including the rooms above and below. This average was within plus or minus one degree of the actual temperature in any of the surrounding space. Between approximately 10:00 p. m. and 8:00 a. m., or from the close of one test to the beginning of the next, the electrical connections were broken and the normal steam supply and control of the building was used.

Thermocouples, read by a precision potentiometer, and mercury thermometers gave the temperature (hereafter called the control temperature) of the air at the breathing line in the center of the room where the thermostat was located, the temperature at various points between the floor and the ceiling, the temperature of the floor, ceiling and partition surfaces, the inside glass surface temperature, and the temperature of the air 3 ft outside of the window. Manometers gave the pressure drop between the air within the room and that outside. Anemometers were used to give the vertical and the two horizontal components of the air movement 3 ft outside of the windows, while an indicating, integrating cup-type anemometer placed at the top of the building with the indicator on the 27th floor gave the miles of wind travel between observations. In most of the tests the above observations were made at intervals of about 1½ to 2 hours. In a few of the tests these observations were made more frequently.

Table 1—Room Characteristics

Room No.	Exposure: Direction	Exposure: Width Ft	Exposure: Window Area Sq Ft	Exposure: Wall Area Sq Ft	Length Corridor to Window	Transmission Loss for Entire Exposed Wall and Glass Btu/Hr/Deg. F Temp. Diff. Based on the Guide 1933
705	N. E.	8.48	40.75	42.62	9.75	64.0
707	N.W.	9.08	38.24	51.26	11.19	63.6
726	S.W.[a]	10.28	40.68	60.21	23.83	69.6
1826	S.W.	9.25	40.75	50.59	15.07	66.5
2703	N.E.	10.04	40.87	57.87	20.05	69.2
2706	N.E.&N.W.	10.33	79.66	123.14	10.33	139.5
2715	N.W.	9.46	37.91	55.09	11.74	64.8
2718	S.W.	10.42	40.62	61.81	20.04	70.4

[a] South 39° 30′ West.

Test Results

Figs. 1 to 5, respectively, give the results of a test on February 6 in five rooms: 726, 1826, 2718, all having approximately equal exposures on Third Avenue or the southwestern side of the building, Room 2715 on the Grant Street or northwestern side of the building, and Room 2703 on Fourth Avenue or the northeastern side of the building. During this test all electrical input ob-

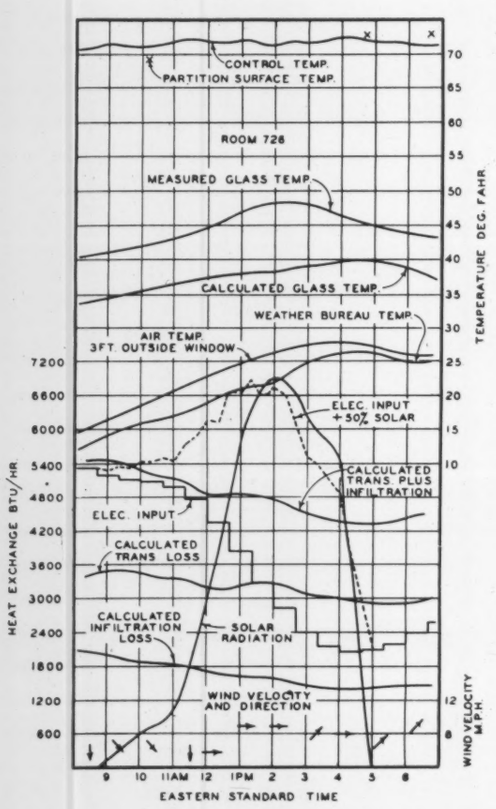


Fig. 1—Log of test on February 6, 1934 in Room 726, having a southwestern exposure

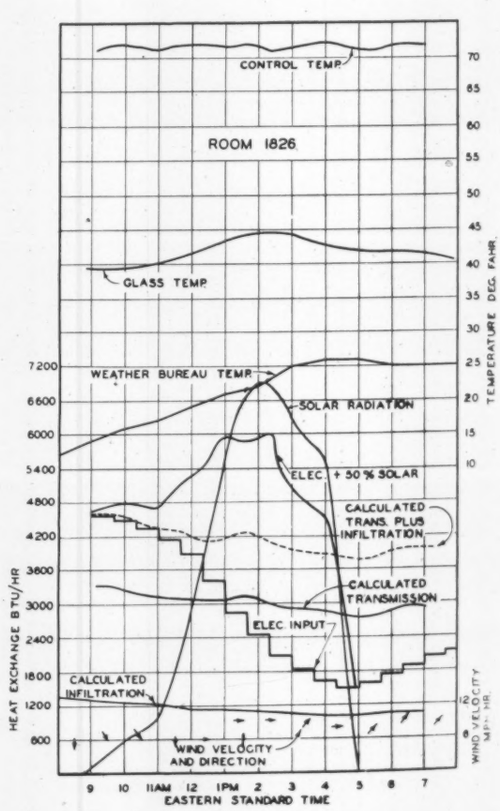


Fig. 2—Log of test on February 6, 1934 in Room 1826, having a southwestern exposure

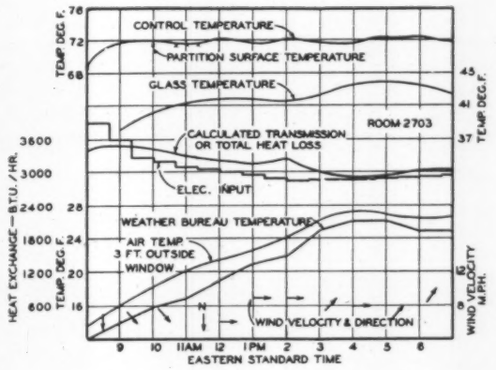


Fig. 3—Log of test on February 6, 1934 in Room 2703, having a northeastern exposure

servations were made at 40 min intervals, and all other observations were made more frequently than was usually the case. The observed electrical input, the observed solar radiation incident upon the glass, and the sum of the electrical input plus 50 per cent of the solar radiation are plotted. The observed wind velocity and direction as given by the Weather Bureau are also plotted. The temperatures reported by the Weather Bureau, and in some instances those observed 3 ft outside of the window, the inside glass surface temperature and the room control temperature are also given. The calculated transmission heat loss, based upon the temperature 3 ft outside of the window, the calculated infiltration loss based upon the pressure drop through the window, and the sum of transmission and infiltration losses, or the total calculated heat loss from the room, are also plotted.

A log of weather conditions and the heat supplied to the several rooms, in Btu per average hour of test and per average degree temperature difference between the inside and outside air, for each test period are plotted in Fig. 6 for all tests from January 18 to March 29. In the upper section of the chart the wind velocity as reported by the Weather Bureau during the test period of each day is plotted, and its prevailing direction during the same period is indicated by an arrow. The precipitation, measured in inches of water, is also given for the 24 hour period ending at 8:00 a. m. E.S.T., the following morning. The symbol *T* represents an undetermined trace of precipitation. The average outside air temperature during the period of each test and the mean 24 hour temperature as reported by the Weather Bureau are indicated by + and □, respectively, in the next lower section of the chart. The temperature range for each daily test period is indicated by a separate curve. In the eight lower sections of the chart the average heat requirement for the test period is given in Btu per hour per degree temperature difference between the temperature of each room and the outside Weather Bureau temperature. In each case the electrical heat input is indicated by an *x*, while for those rooms affected by solar radiation through the window, the sum of the electrical heat input and 50 per cent of the solar radiation incident against the glass of the windows is indicated by *o*.

The daily average heat loss in Btu per hour per average degree temperature difference between the inside temperature and the outside Weather Bureau recorded temperatures are plotted in Figs. 7 and 8 against the average wind velocity in miles per hour for each test period for the different tests in Rooms 705 and 2703, respectively. Both rooms are on the Fourth Avenue or northeastern side of the building and, therefore, subject to a negligible amount of solar radiation. The prevailing wind direction during each test is indicated.

The average hourly electrical heat input is plotted against outside temperature for all tests in Rooms 707, 726 and 2703 in Figs. 9, 10 and 11, respectively. Room 2703 on the northeastern side received practically no

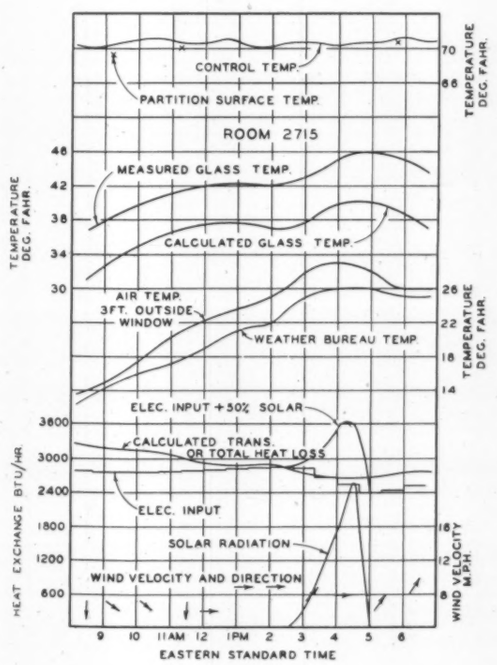


Fig. 4—Log of test on February 6, 1934 in Room 2715, having a northwestern exposure

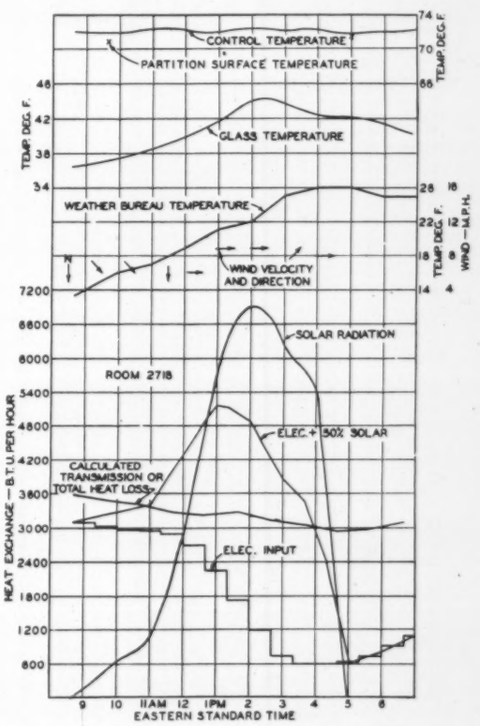


Fig. 5—Log of test on February 6, 1934 in Room 2718, having a southwestern exposure

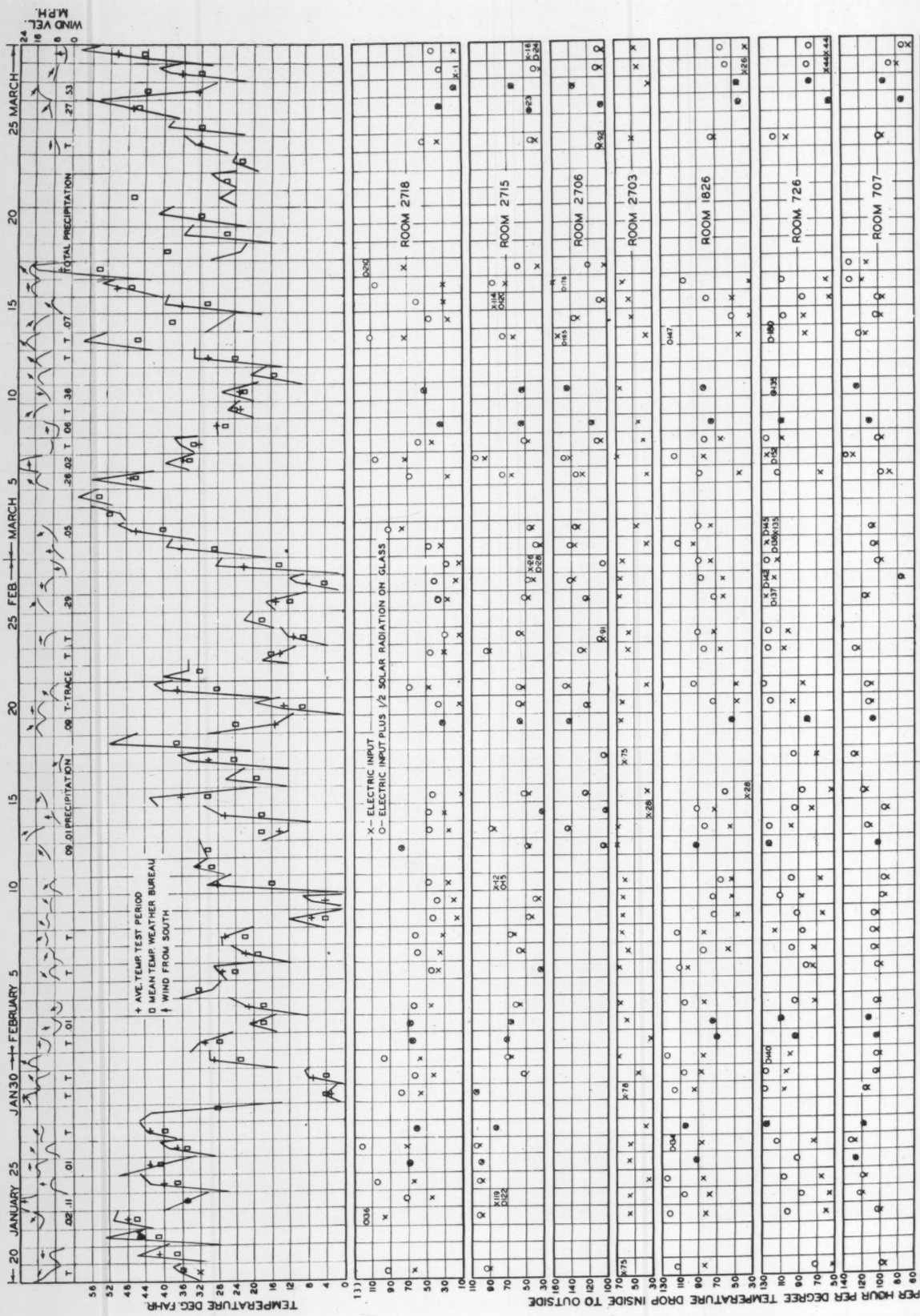


Fig. 6—Log of weather conditions and daily heat requirements in Btu per average hour of each test period per average degree temperature difference between the inside and outside, during each test period, for the eight rooms from June 19 to March 29

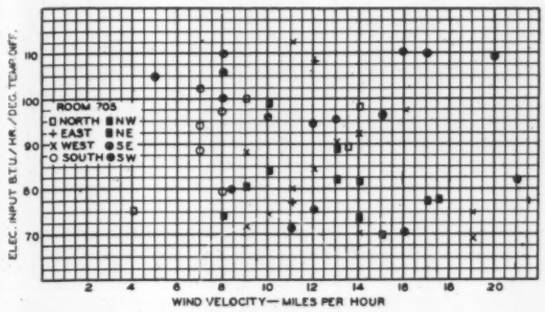


Fig. 7—Relation between wind velocity and direction and electric heat input for all tests in Room 705

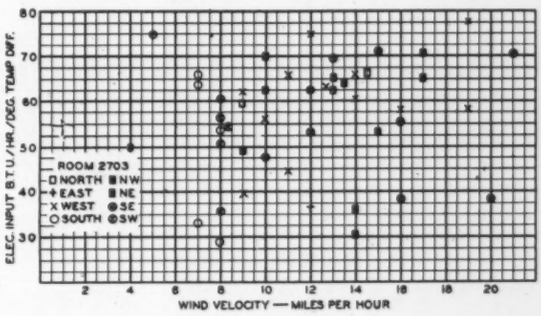


Fig. 8—Relation between wind velocity and direction and electric heat input for all tests in Room 2703

solar radiation, Room 707 on the northwestern side received very little, while Room 726 received considerable solar radiation. Points resulting from tests when there was no solar radiation are indicated as *x* in Fig. 10, for Room 726.

During the period of the study, the control of the steam supply to the building and observations on the rate of consumption were made available to the Research Laboratory staff. In Fig. 12 the daily steam consumption per degree day in the building is plotted against degree days, while in Fig. 13, the pounds of steam per hour per degree day are plotted against degree days. In Fig. 14 the average daily steam pressure in pounds per square inch maintained on the building side of the reducing valve is plotted against the daily mean temperature and the resulting degree days. Since the building is heated by a differential system, the steam pressure at this point is not very different from that existing throughout the system, and the steam temperature corresponding with these pressures as given in the chart is approximately the steam temperature throughout the system, or the temperature of the direct surfaces of the heating units.

Discussion of Results

The data plotted in Figs. 1 to 6 show marked variations in the hourly heat required per degree temperature difference to maintain the desired control temperatures in the several rooms, depending upon the elevation of the room above the street level, the solar radiation effect, and other factors.

Chimney Effect of the Building

Figs. 1, 2 and 5, giving complete logs of the test on February 6 in Rooms 726, 1826 and 2718, all having approximately the same exposure on the 7th, 18th and 27th floors, show calculated heat losses based upon transmission and infiltration, and heat inputs, including electrical input and electrical input plus 50 per cent of the estimated solar radiation incident on the glass of any room, which decreases in magnitude with elevation in the building. The average calculated heat requirement, the average electrical input and the average electrical input plus 50 per cent solar radiation for these rooms are, respectively 4747, 3677 and 4950 for Room 726; 4114, 2758 and 4154 for Room 1826; and 3215, 1784 and 3174 for Room 2718.

The average values for all tests in the several rooms given in Table 2 also indicate that the calculated heat requirements, electrical inputs, and electrical inputs plus one-half of the solar radiation decrease in magnitude with elevation in the building. This is particularly true when rooms of the same exposure, but on different floors, are considered. To a lesser extent this is also true when all rooms on the different floors are considered. These variations in heat requirement with elevation in the building are obviously due to the chimney effect of the

Table 2—Relation Between the Calculated and Observed Heat Requirements for the Several Rooms Studied in the Grant Building

A	B	C	D	E	F	G	H	I	J
Room No.	Calculated Trans. Btu/Hr/Deg. F Temp. Diff.[b]	Calculated Infiltration Btu/Hr/Deg. F Temp. Diff.[c]	Calculated Transmission and Infiltration Btu/Hr/Deg. F Temp. Diff.	Meas. Elec. Input Ave. All Tests Btu/Hr/Deg. F Temp. Diff.[d]	Meas. Elec. Input Calc. Trans. and Infil. (E/D) x 100=Per Cent	Range Elec. Input All Tests Btu/Hr/Deg. F Temp. Diff.	Elec. Input Plus 50 Per Cent Solar Rad. Ave. All Tests Btu/Hr/Deg. F Temp. Diff.[d]	Meas. Elec. Input Plus 50 Per Cent Solar Radiation Cal. Trans. and Infiltration H/D x 100=Per Cent	Range Elec. Input Plus 50 Per Cent Solar Radiation Btu/Hr/Deg. F Temp. Diff.
705	64.0	32.0	96.0	88.0	91.7	70–112	88.0	91.7	70–112
707	63.6	34.0	97.6	104.3	106.9	65–128	107.5	110.1	65–134
726	69.6	34.2	103.8	87.8	84.6	44–135	106.5	102.6	54–180
1826	66.5	23.2	89.7	67.6	75.4	26–105	88.7	98.9	45–147
2703	69.2	0	69.2	55.0	79.5	28–78	55.0	79.5	28–78
2706	139.5	0	139.5	122.2	87.6	91–160	126.0	90.3	93–178
2715	64.8	0	64.8	59.6	92.0	12–119	63.3	97.6	15–122
2718	70.4	0	70.4	41.3	58.7	1–98	64.5	91.6	18–210
726 Ave. 1826 2718					79.2			97.7	
Ave. 705 2703					85.6			85.6	
Ave. 707 2715					99.4			103.8	
7th Floor Ave.					94.4			101.4	
27th Floor Ave.					79.4			89.7	
Ave. All Rooms					84.5			95.3	

[b] Col. B—Based on The Guide 1935 heat transmission coefficients and 50 F temperature difference.

[c] Col. C—Based on the average observed pressure difference through the window for all tests, the observed relation between the pressure difference and air flow, and a 50 F temperature difference.

[d] Col. E and H—Average of all test values from Fig. 6.

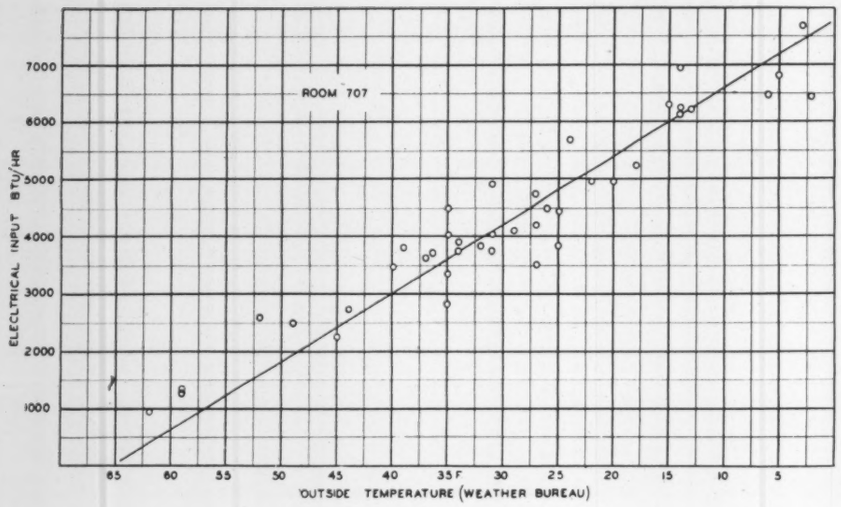


Fig. 9—Relation between outside temperature and electric heat input for all tests in Room 707

building, resulting in excessive infiltration in the lower floors, lesser infiltration on the 18th floor, and exfiltration only in the average case for rooms on the 27th floor. This effect is reflected in the calculated values for the reason that the average measured pressure difference between the inside and outside of the window as observed in the different tests was used in calculating the infiltration. The chimney effect of the building and the resulting infiltration and its effect on the heat requirements of the building will be discussed further in a future laboratory report.

Solar Radiation

The tremendous effect which solar radiation has on the heat requirements of rooms with a sunny exposure is shown by the plotted results for the test on February 6 in Rooms 726, 1826 and 2718, Figs. 1, 2 and 5, respectively. The exposure of these rooms faces Third Avenue or the southwest. On this day, the sun's rays were first intercepted by the windows of these rooms at 8:42 a. m. and left them at 5:00 p. m., reaching a maximum value at about 2:00 p. m. The curves give the intensity of the solar radiation impinging on all the glass surface of these windows in Btu per hour. Since the three rooms had the same exposure and glass area, the curves in the three figures are identical.

Obviously, all of the solar energy striking the glass is not transmitted, and of that part which is transmitted through the glass and absorbed by the floor, furniture and other surfaces, only a part is immediately given back to the air in the room. In another laboratory report[2] dealing with the cooling effect in some of these same rooms, it was found that during the summer cooling period approximately 50 per cent of the solar radiation impinging on the glass became effective in the cooling load during the cooling period from 7:00 a. m. to 4:00 p. m. E.S.T. For this reason, 50 per cent of the solar radiation impinging on the glass is added to the electrical input in establishing the curve, *electrical input plus 50 per cent solar radiation.*

It will be observed that for Room 2703, Fig. 3, which received no solar radiation during the test period, the electrical input correlates well with either the variation in outside temperature, or the calculated heat loss as plotted. For the other rooms which received solar radiation, the electrical input is considerably affected by the heat gain from solar radiation. However, the reduction in electrical input does not follow closely the increase in solar radiation, but lags considerably behind. Thus, for Rooms 726, 1826 and 2718, while the maximum rate of solar radiation impinging on the glass took place at 2:00 p. m., the minimum rate of electrical input resulting therefrom took place at times ranging from 3:00 to 5:00 p. m. for the different rooms. The curves for these rooms giving the electrical input plus 50 per cent of the solar radiation impinging against the glass, when compared with the curves for calculated heat loss, show that the combined heat input is excessive during the time when the solar radiation is effective and too small after the solar radiation ceases to be effective, showing that a considerable part of the solar radiation entering the room was stored as heat in the floor and furniture, to be given back to the air in the room at a later time.

The curves in Fig. 4 show less solar radiation effect in Room 2715 than in Rooms 726, 1826 and 2718. The time when the sun became effective in this room was later, the period of effectiveness was shorter, and the maximum intensity of the radiation was less. However, a similar lag between the time of the radiation and its effect on the heating load is shown.

It is apparent from the curves in Figs. 1 to 5, that solar radiation has a considerable effect on the heat requirements of rooms affected, that this effect is much less than the total rate of radiation impinging against the window, and that a considerable lag exists between the time of maximum solar radiation against the window and its effect on the heat load of the room. The effect of solar radiation on the heating requirements of rooms having sunny exposures is further made apparent by the average hourly heat requirement of the several rooms for all tests plotted in Fig. 6. Rooms 705 and 2703, facing Fourth Avenue or the northeast, which received only a negligible amount of solar radiation in the early morning, show much less variation in the heat requirement than rooms receiving solar radiation.

It is apparent that on days during which solar radiation is a factor, electrical input is considerably less. However, the sum of electrical input plus 50 per cent of the solar radiation gives a variable value frequently considerably in

[2]Cooling Requirements of Single Rooms in a Modern Office Building, by F. C. Houghten, Carl Gutberlet and Albert J. Wahl. A.S.H.V.E. Journal Section (*Heating, Piping and Air Conditioning*), April 1935, p. 207.

excess of the electrical input for the same room on days when there is no solar radiation, indicating that great difficulty would be involved in properly evaluating the effect of solar radiation on the heat requirement of a given room on a given day. Obviously, solar radiation cannot be taken into account in estimating the maximum heating requirement of a given room or building, since the advantage can only be depended upon when the sun is shining. The sun effect is, however, a considerable factor in reducing the total heat required in such rooms during the heating season.

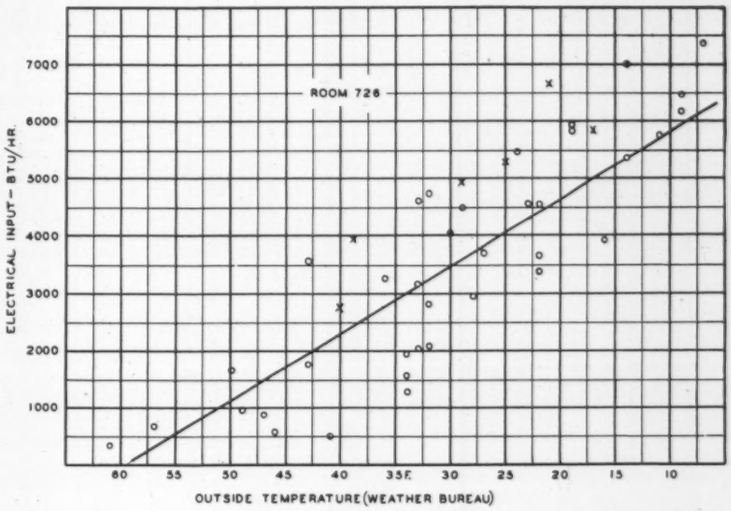


Fig. 10—Relation between outside temperature and electric heat input for all tests in Room 726

Variation in Heating Load with Outside Temperature

The electrical input required to maintain the desired temperature in Rooms 707, 726 and 2703 is plotted against outside Weather Bureau temperature in Figs. 9, 10 and 11. The individual test points plot more closely to the average curve for Room 2703, receiving no solar radiation, and spread out most for Room 726, Fig. 10, receiving the most solar radiation, where test points for days when there was no solar radiation are plotted as *x*. These points indicate a separate curve for such days, requiring a greater heat demand per degree temperature difference between the outside and inside, as should be expected, since the effect of solar radiation is to reduce the heat requirement by variable amounts, depending on the intensity and continuity of sunshine. The curve for all tests in Room 707, where solar radiation had a small effect, and a curve for Room 726, when there was no solar radiation, intersect the outside temperature axis between 64 and 65 F outside temperature, indicating that for these conditions no heat is required with approximately 65 F outside temperature. Room 2703 on the 27th floor, which received no solar radiation, gives a curve intersecting the outside temperature axis at approximately 60 F. This difference appears to be due to the chimney effect of the building. On the 7th floor there was always a pressure difference from the outside to the inside, accompanied by a high infiltration rate which increased with lower outside temperatures, while on the 27th floor there was usually a pressure drop from the inside to the outside, resulting in exfiltration, causing no heat loss. All test points for Room 726, irrespective of solar radiation, give a curve intersecting the outside Weather Bureau temperature at about 60 F, while the lower limits of the points for this room would give a curve intersecting the outside temperature at about 50 F, showing that this room, based on an average of all tests irrespective of solar radiation, required heat when the outside temperature was about 60 F, and for most sunny days it did not require heat when the outside temperature reached 50 F. The data plotted in Figs. 9, 10 and 11 show the usually accepted straight line relationship between heat requirement and temperature difference.

Accuracy with Which Heat Requirement May Be Estimated

Further inspection of the heating requirements for the various tests and for the several rooms plotted in Fig. 6 shows variation which cannot be accounted for by solar radiation, elevation in the building, wind velocity, wind direction, or precipitation. As examples, the individual heat requirement points for Rooms 705 and 2703, where solar radiation was not a factor, show a range from 70 to 112 and from 28 to 78, respectively. It was shown previously that these variations could not be accounted for by wind velocity, direction, or precipitation. They appear, therefore, to be variations resulting from other uncontrolled factors in the study.

Since an attempt was made, insofar as possible, to take into account all variable factors, this variation would

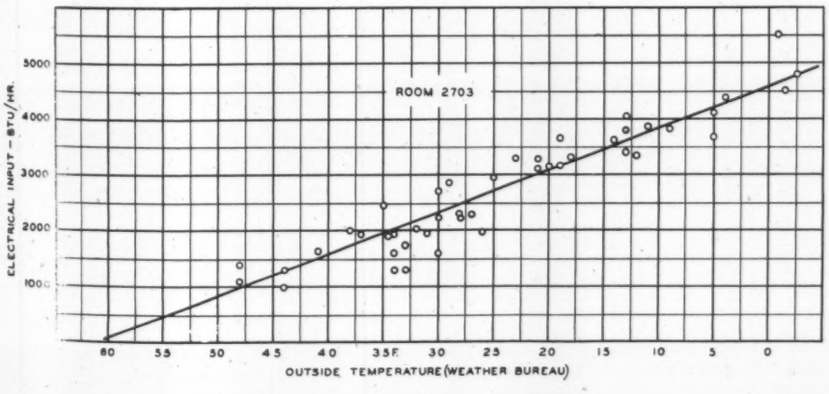


Fig. 11—Relation between outside temperature and electric heat input for all tests in Room 2703

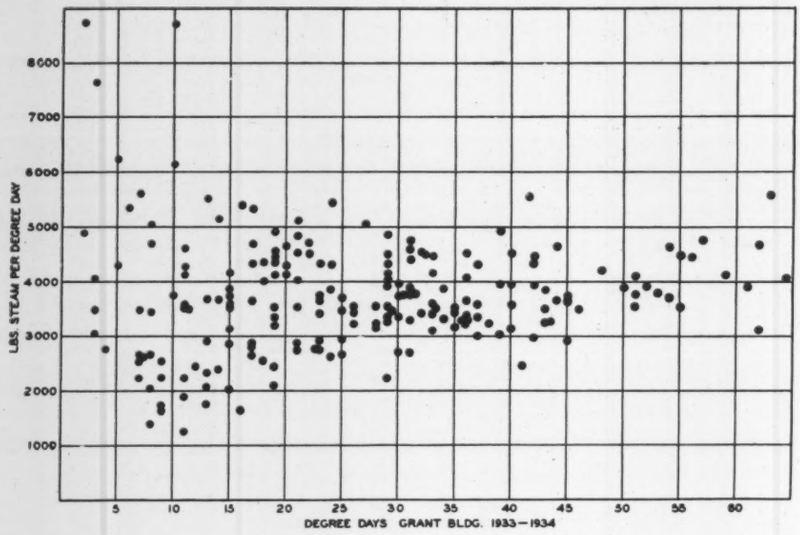


Fig. 12—Relation between degree days and pounds of steam consumed per degree day in the Grant Building during the 1933-34 heating season

seem to be uncontrolled. In other words, it would indicate that taking all factors of exposure, types of construction, infiltration, etc., into account, the heat requirement of a given room still varies considerably from day to day. Any effect which variation in wind may have apparently comes within these limits and could not be evaluated from the findings of the study. In observing the heat requirement values for the different rooms, plotted in Fig. 6, there appears an apparent variation with time. Thus, Rooms 1826, 2715 and 2718 seem to show a higher heat requirement for the same temperature difference during the first few weeks of the study. No reason for this variation with time is apparent. It might possibly be accounted for by variations in moisture content of the building structure, or other factors beyond the attempted control of the study. The heat requirement, including the electrical input and 50 per cent of the solar radiation, expressed as a percentage of the calculated heat requirement in column I, Table 2, shows unaccountable variations for certain rooms. Room 707 shows a high percentage, while Room 2703 shows an unusually low value, which was consistent for all tests, indicating a low heating requirement. In part, this difference may be due to high infiltration in the case of 707, which is in the lower part of the building, and no infiltration in the case of 2703. This effect has, however, been taken into account in calculating the infiltration loss in column C.

Relation Between the Steam Required in the Entire Building and Outside Weather Conditions

In Fig. 12 the steam required per degree day is plotted against degree days for the 1933-1934 heating season. It is observed that less variation in steam consumption exists for colder weather. When the number of degree days per day is less than 20, a wide variation in the points is apparent. However, when the same points are divided by the number of hours of steam supply during any individual day and again plotted against degree days per day in Fig. 13, a very definite relationship is shown, giving a fairly constant rate of approximately 200 lb of steam per hour per degree day for days of 50 or more degree days, with an increasing rate of hourly steam consumption per degree day for milder weather. This further indicates the commonly accepted fact that heating requirement is more efficiently supplied in cold weather than in mild weather, and that the lack of efficiency be-

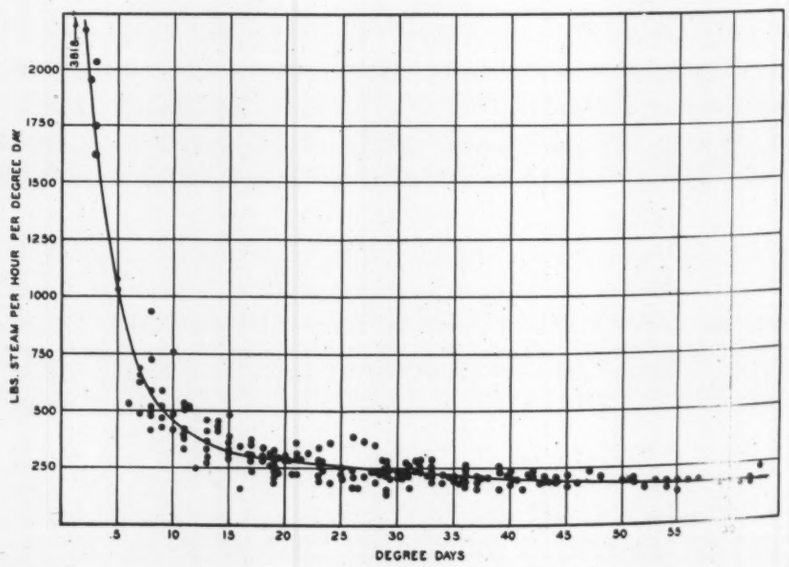


Fig. 13—Relation between degree days and pounds of steam consumed per hour per degree day during the 1933-34 heating season

comes pronounced as the outside temperature approaches 65 F.

The data plotted in Fig. 14, giving the average daily steam pressure in pounds absolute per square inch required for varying daily mean temperatures and degree days are of interest. The differential vacuum system in the Grant Building requires a pressure difference between the steam on the building side of the reducing valve and that maintained by the vacuum pump of

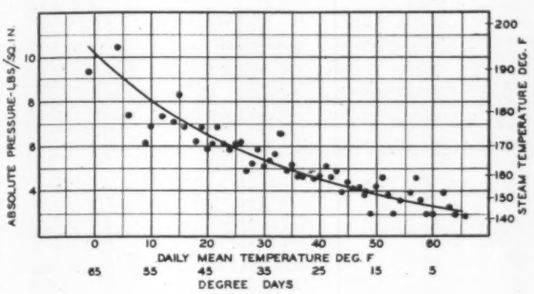


Fig. 14—Relation between the daily mean temperature and the resulting degree days and the average daily steam pressure in pounds absolute required in the heating system of the Grant Building during the 1933-34 heating season

approximately 1 in. mercury column. Therefore, the pressure of the steam and the resulting steam temperature given in Fig. 14 indicate fairly accurately the mean heating unit surface temperature throughout the building and its relation to outside weather conditions.

Summary and Conclusions

1. This report gives data on the electrical input required to heat eight rooms located on three floors and having three different exposures in the Grant Building, Pittsburgh, to approximately 70 F.

2. The study briefly indicates a considerable reduction in heating requirement of the rooms on upper floors below that required for rooms on lower floors due to the chimney effect of the building. This subject will be discussed more thoroughly in another report from the Laboratory.

3. It is shown that the sun effect on the heating requirement of rooms subject to solar radiation is considerable and varies from no effect on cloudy days to a very large effect on days of more intense radiation. There is considerable lag between the time of maximum solar radiation effect and the resulting reduction in the heating load for any particular room. The percentage of solar radiation impinging against the glass of the window in a given room, which is effective in heating that room during the usual heating period of the day, varies considerably, but averages about 50 per cent.

4. The relation between heating requirement and temperature difference between the inside and outside air is indicated to be a straight line relationship, no heat being required when the outside temperature is above 65 F for the lower floors of the building, and at a slightly lower temperature for rooms in the upper part of the building where infiltration is not a factor. Solar radiation reduces the outside temperature at which heating is required on sunny days.

5. For best conditions of control in rooms where solar radiation was not a factor during the test period, considerable variation in the heat requirement per degree temperature difference was observed which could not be accounted for by wind velocity or direction, or by precipitation.

Coal Classification Committee Develops Size Designation Test

A method for designating the size of coal from its screen analysis was approved for recommendation as a tentative standard method to the *American Society for Testing Materials* at a meeting of the Sectional Committee on Classification of Coal. The *A.S.T.M.*, under the procedure of the *American Standards Association*, is sponsor of the Sectional Committee of which A. C. Fieldner, chief engineer, Experiment Stations Division, U. S. Bureau of Mines, Washington, D. C., is chairman and J. D. Battle, *National Coal Association*, Washington, D. C., is secretary.

The method covers the designation of coal sizes from the results of screen analysis tests of samples taken to represent the condition of the coal as sold. The size designation of a coal is expressed in terms of the lower and upper limiting screens between which more than 80 per cent of the sample is retained by actual test. A series of testing screens is included. The lower limiting screen is the largest screen in the series through which passes a total of less than 15 per cent of the sample. The screen defining the upper limit is the smallest screen of the series, upon which is retained a total of less than 5 per cent of the sample.

A minor revision in the Tentative Specifications for Classification of Coal by Rank (D 388-35 T) was also voted by the Committee, involving the substitution of an agglomeration test for the agglutinating value test to determine caking properties. The agglomeration test has the advantage of simplicity since no special apparatus is needed. The agglomerating or caking characteristics of coals are judged by examination of the residue in the platinum crucible incident to the volatile matter determination.

Dudley Medal Awarded to H. C. Mann by A.S.T.M.

Announcement has been made of the selection of H. C. Mann, senior materials engineer, Ordnance Dept., U. S. Government, Watertown Arsenal, to receive the Charles B. Dudley Medal for 1936. This medal commemorates the name of the first president of *A.S.T.M.* and is awarded to the author of the paper presented at the preceding annual meeting which is of outstanding merit and contributes original research in engineering materials.

The paper entitled, The Relation Between the Tension Static and Dynamic Tests, presents data and conclusions from an investigation which led to the discovery of a correlation between static and dynamic tests.

Formal presentation of the medal to Mr. Mann was made during the 1936 annual meeting of the society in Atlantic City, N. J., June 29-July 3.

Relative Absorption of the Various Sections of a Round Heating Boiler

By A. J. Johnson* (*MEMBER*) and P. A. Mulcey** (*NON-MEMBER*)
Primos, Pa.

AN understanding of the relative value and exact part played by each individual section of a vertical or round heating boiler is of considerable importance for a number of reasons. A knowledge of the increased efficiency and output which may be expected as the result of adding one or more intermediate boiler sections is of definite value both from an economic and engineering viewpoint. The proportion of radiant heat absorbed by the furnace side walls and by the crown sheet is useful in determining the proper fuel bed depths to be carried with various solid fuels. The effect of forcing flame to impinge upon boiler side walls gives definite information concerning the practical value of furnace baffles.

A study of heat absorption from fuels, such as coal and coke, which have compact incandescent fuel beds, as compared with liquid or gaseous fuels with which heat is generated, without the use of a fixed fuel bed, is also of interest.

Many other aspects of an investigation of this type also prove useful in designing and proportioning heaters. For this reason every effort was made to secure results which would be typical of actual conditions such as might be repeatedly encountered in the field.

Description of Apparatus

The tests were conducted in a 25 in. five-section cast iron boiler. The firepot section, approximately 2 ft in diameter, was tapered slightly inward at the top with an overall depth of 24 in. The flue passes within the intermediate sections were approximately 3 in. deep so that the initial gas velocity was several times greater than in the furnace. The absorption surfaces of the several sections were:

Firepot Section	13.5 sq ft
Crown Sheet	11.3
Each Intermediate (2 Sections)	11.0
Dome Section	7.0
Total	53.8 sq ft

Connecting nipples between each section as well as between the furnace side wall section and crown sheet were filled with lead so that water could be passed through each unit separately and its heat rise measured.

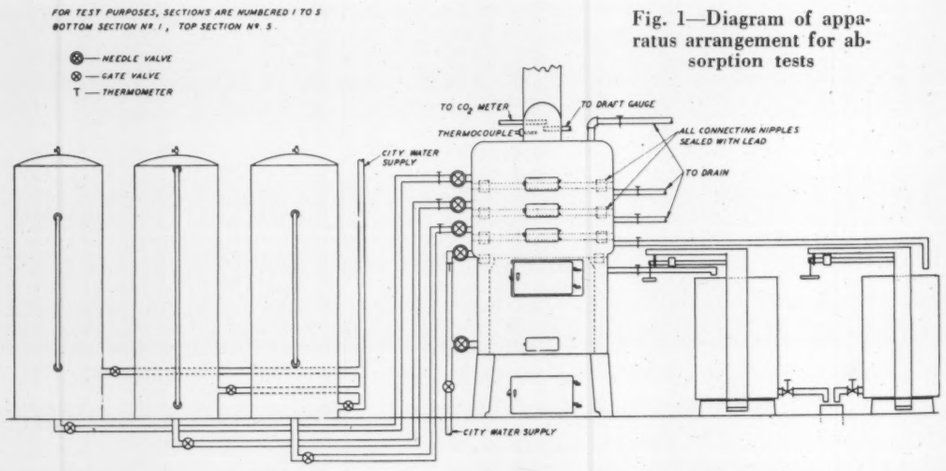


Fig. 1—Diagram of apparatus arrangement for absorption tests

The outlet temperature from each section was graduated approximately from 120 F in the firepot section to 160 F in the top section in order to simulate normal conditions in which the hottest water would be at the top of the heater. Even with this precaution, however, boiler efficiencies were somewhat higher than normal.

Various fuels were used, including anthracite, by-product coke, petroleum coke, oil, and gas. Oil was fired in a rotary type burner, and the gas was consumed in a conversion type burner.

Hand fired tests were all operated as cycle tests with all the usual readings and measurements for determining the efficiency and conditions of combustion. The automatic equipment tests were 8 hour, running start tests.

A graphic picturization of the testing layout is shown

*Director, Anthracite Institute Laboratory.

**Assistant Director, Anthracite Institute Laboratory.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1936, Buck Hill Falls, Pa.

in Fig. 1, whereas the log sheets given in the discussion of this paper show the draft and fuel bed conditions of the various tests.

Average Absorption with All Fuels

As expressed in terms of percentage (with the total output of the heater as 100 per cent),[1] the absorption of each section remained surprisingly constant with each of the various fuels. Table 1 and Fig. 2 show the general average results with all fuels, whereas Table 2 gives individual results with each fuel.

Table 1—Absorption of Each Section Averaged for All Fuels

Side Wall Section	56.9%
Crown Sheet Section	17.9%
1st Intermediate	13.1%
2nd Intermediate	7.9%
Dome Section	4.2%
Total Boiler Output	100.0%

The only plausible explanation of the similarity of the distribution of heat absorption throughout the boiler when using solid and fluid fuels would seem to be that a sufficient amount of heat is radiated from the glowing mass of burning oil or gas to balance the radiant heat from a bed of solid fuel.

To substantiate this conclusion it will be noted that the results with an underfeed stoker without a baffle were of an entirely different character, in that the absorption of the side walls was lower than the average by some 17 per cent, whereas the crown sheet absorbed 11 per cent more than the all-fuel average. In this type of stoker the fuel bed is relatively flat and is constructed in such a manner that nearly all radiant heat will be transmitted to the crown sheet rather than at an oblique angle to the side walls. The flame from the stoker also differs from oil and gas in that it is probably merely the luminous products of combustion rather than the actual burning mass of an appreciable weight of fuel.

Table 2—Relative Amount of Heat Absorbed by Each Boiler Section

(Expressed as Per Cent of the Total Heater Output)

Fuel \ Boiler Part	No. of Tests	1 Side Walls	2 Crown Sheet	3 First Intermediate	4 Second Intermediate	5 Dome	Total
Anthracite [a]							
Egg	6	56.8	17.0	13.1	8.3	4.8	100.0
Stove	6	57.9	16.9	13.0	7.8	4.4	100.0
Chestnut	10	58.1	17.8	12.5	7.8	3.8	100.0
Pea	6	56.4	17.4	13.9	8.0	4.3	100.0
Buckwheat [b]	5	57.0	18.0	13.8	7.5	3.7	100.0
Ave. [c]		57.3	17.4	13.2	7.9	4.2	100.0
Anthracite Stoker [d] with Baffle	2	55.4	18.4	13.2	8.5	4.5	100.0
Anthracite Stoker [e] without Baffle	4	38.9	30.1	15.8	9.4	5.8	100.0
Petroleum Coke	6	57.8	14.9	12.9	8.7	5.7	100.0
By-Product Coke	6	55.2	16.5	14.0	9.4	4.9	100.0
Oil Burner	4	52.0	21.8	15.1	7.7	3.4	100.0
Gas Burner [f]	4	64.1	18.6	10.0	5.2	2.1	100.0
Ave. [g]		56.9	17.9	13.1	7.9	4.2	100.0

[a] Hand fired only.
[b] Induced draft.
[c] This average includes hand fired anthracites only.
[d] With 21 in. baffle 10 in. above firepot.
[e] Not included in average because of different nature.
[f] Including refractory brick.
[g] Includes all fuels except stoker without baffle.

[1] Averages in this section refer to per cent of actual total heat absorbed by boiler and are not to be confused with efficiency results which would of course refer to per cent of input.

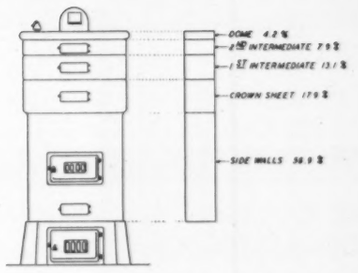


Fig. 2—Round heater with each section drawn in proportion to amount of heat (average) which it absorbed

Proportion of Heat Absorbed

In the first or side wall section of the boiler, absorption by conduction, radiation, and convection can and probably does occur simultaneously. In the crown sheet, only radiation and convection are possible (unless a flame from actual burning fuel impinges upon the crown sheet). In all other heating surfaces, absorption is limited to that from convection.

Absorption by Conduction

In oil and gas furnaces absorption by conduction is insignificant, whereas even in burning solid fuels these tests indicated that it is a negligible quantity. The *point to point* contact of solid fuels against the side walls does not offer any large or very effective surfaces for transmission of heat by conduction. The conductivity of fuels is relatively low, so that there is little likelihood of a free flow of conducted heat horizontally through a fuel bed, and accumulations of ash and dead fire against the side walls act as an insulator.

While it is difficult and impractical to secure actual measurements which would segregate radiated and conducted heat to the side walls, calculations nevertheless substantiate this conclusion, since little unaccounted absorption remains after deducting the calculated radiant and convected heat.

Absorption by Radiation

Assuming a uniform temperature over the surface exposed vertically to the furnace walls and to the crown sheet, the calculated radiation to the firepot or side walls would be from 3.5 to 6 times greater than that to the crown sheet. This ratio would, however, be considerably reduced if, as is undoubtedly the case, the upper part of the fuel bed is hotter than the lower. Substantiating these calculations, in all hand fired tests, the actual measured absorption of the side walls was from 2.4 to 3.6 times greater per square foot than that absorbed in the crown sheet section.

This indicates that most of the heat absorbed by the side walls is by radiation rather than by convection, since if the latter were the case the heat absorbed per square foot would probably be greater for the crown sheet because of the sweep of hot gases across its surfaces.

Absorption by Convection

As compared with radiation, convection is extremely difficult to calculate because of such variables as gas

velocity, etc. Therefore, in order to determine the amount of heat absorbed in each section by convection, the total heat actually absorbed in the tests was plotted as measured. Radiation was then calculated and plotted on the same curves. Neglecting conduction, and it is probably negligible for reasons which have been explained, the difference between this actual total absorption and the absorption by radiation will represent convection. The heat absorbed by each section in typical individual tests is shown in Figs. 3 and 4. Total absorption as measured, radiation and convection as calculated, were plotted as at the midpoints of each boiler section upon abscissae representing square feet of heating surface. As the ordinates represented heat absorbed per square foot, the area of the shaded blocks is proportionate to the heat absorbed by each section. Fig. 3 is a chestnut anthracite test at a rate of 4 lb per square foot per hour, whereas Fig. 4 is the same fuel but at the higher rate of 8.6 lb per square foot per hour.

Economic Value of Added Boiler Sections

Aside from the design standpoint, one of the most important uses of this type of information is a determination of the economic value of added boiler sections. The side walls and at least a combined crown sheet and dome are of course indispensable. It is possible, however, through the addition of one or more intermediate sections, to increase the absorbing power of the boiler to a limit determined only by the economic ratio between the output to be expected and the cost of the added metal.

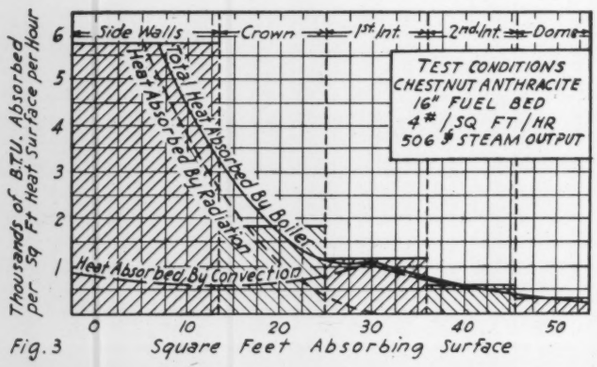


Fig. 3

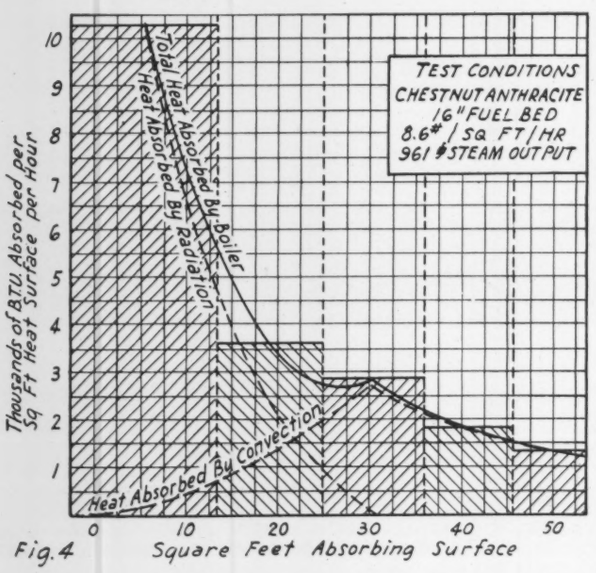

TEST CONDITIONS
CHESTNUT ANTHRACITE
16" FUEL BED
8.6# / SQ FT / HR
961 # STEAM OUTPUT
Total Heat Absorbed By Boiler
Heat Absorbed By Radiation
Heat Absorbed By Convection
Thousands of B.T.U. Absorbed per Sq Ft Heat Surface per Hour
0 10 20 30 40 50
Square Feet Absorbing Surface

Fig. 4

Figs. 3 and 4—Heat absorbed by individual sections showing calculated convection and radiation for one typical test

Recoverable Stack Loss

In the case of the present tests, the sensible heat carried off by the stack gases is in part recoverable by the addition of sufficient sections to reduce the temperature of the gases to about 212 F. This boiler, however, is already equipped with two intermediate sections having corrugated heating surfaces and the addition of further sections can be readily shown to be both uneconomical and impractical. In Table 3 the test (No. 2230-J) with chestnut coal at maximum output shows 9.5 per cent *recoverable* heat in the stack loss. Assuming absorption to be as rapid as in the top section, (and this is not strictly true because of the fact that the curve of absorption flattens out as its abscissa increases in length, as shown in Figs. 3 and 4), to recover this heat would require the addition of about 28 sq ft of absorption surface, increasing the present boiler size over 50 per cent.

Furthermore, at lower outputs this recoverable loss becomes nearly negligible, being less than 2½ per cent at one-half boiler rating.

The problem, therefore, consists of extending curves of absorption, such as Figs. 3 and 4, which can then be used as fair estimates of expected additional heat recovery.

While it is seldom desired to remove a section from an existing boiler, the same curves can of course be used as a basis for estimating probable effect.

Reduction of Side Wall Area and Combustion Space

The great majority of anthracite burning domestic boilers are designed in accordance with the requirements of high volatile fuels for large furnace volumes, to insure complete combustion. It has been demonstrated, however, that anthracite can be burned satisfactorily with practically no combustion space, such as in *Anthracite Institute* horizontal combustion boilers. With anthracite, the chief function of the furnace section is to absorb as much as possible of heat radiated directly from the fuel bed.

The most efficient design for surface-fired or magazine-fed anthracite boilers should therefore include:

1. A firepot of sufficient depth to permit the thickest possible fuel bed consistent with the efficient combustion of the required size, and of sufficient grate area to allow the maximum combustion space consistent with the proper feeding of fuel.
2. Flue passes having a maximum of surface exposed to gases at relatively high velocity without unduly increasing the pressure drop or the size of the unit.

Small Furnaces for Stokers

Mention has already been made of the fact that when using anthracite, stoker fired, the side walls are of minor importance as compared with the crown sheet. In view of the sootlessness of anthracite, together with the fact that it is largely a pure carbon rather than a hydrocarbon, it would seem obvious that combustion space over the fire was of minor importance as compared with

radiant surfaces directly above the fire, together with well designed flue passes.

This conclusion has in fact been substantiated by a number of tests within the *Anthracite Institute Laboratory,* in which it has been found possible to burn anthracite in an underfeed stoker set at a distance below the crown sheet just sufficient to prevent actual contact with the incandescent fuel. Outputs as high as 350,000 Btu per cu ft of combustion space have been obtained in this laboratory (see report 2057). The ideal stoker furnace therefore should have just sufficient volume to permit lighting of the fire, feeding of ashes into the ashpit, and giving access to replaceable parts.

According to these specifications, the 25 in. boiler in which the relative absorption tests were conducted could be reduced in height by nearly 30 per cent by cutting the furnace height to about 6 in. Or, without altering the external dimensions, but replacing the present unnecessary furnace space by additional intermediate sections, the convection absorption surface could be more than doubled. This would give the stoker used in the tests an output about 10 per cent higher, with correspondingly higher efficiency.

Conclusions

A number of other variables which might affect the absorption, distribution, and efficiency of a boiler are also of interest. Included in these would be such comprehensive studies as the relation between draft and efficiency; fuel bed depths with various outputs; and the correct design of furnace baffles when using stoker fired or fluid fuels. However, as each of these subjects is of sufficient complexity to warrant a more complete discussion than can be given as a part of this paper, the necessary detailed tests were not included in this investigation. However, in view of the importance of such subjects, together with the scarcity of accurate knowledge concerning these various phases of combustion, it is hoped that other investigators will continue this type of work to provide additional data.

Table 3—Results of All Tests Showing Conditions of Fuel Bed Depth, Draft, Output and Per Cent Output of Each Section

Test No.	Fuel Bed Depth In.	Stack Draft	Duration Hours	Coal Per Hour	Overall Eff.	Total Output Btu	Output, Btu Per Hour					Output, Per Cent of Total					Absorption, Per Cent of Total Input					
							Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Stack[a]
SECTION I. ANTHRACITE—EGG SIZE																						
2230-BD	16	0.03	8.580	19.36	64.8	1,462620	170220	99700	28700	21820	12600	7400	58.60	16.85	12.82	7.39	4.34	38.00	10.92	8.31	4.79	2.82	11.00
2230-BE	16	0.06	8.330	21.56	70.1	1,712654	205000	120000	34200	25550	15660	9530	58.60	16.67	12.42	7.64	4.67	41.10	11.68	8.70	5.35	3.27	12.10
2230-BF	16	0.09	8.250	22.32	71.5	1,785047	216380	121500	37750	28700	18350	10080	56.20	17.42	13.25	8.48	4.65	40.20	12.46	9.47	6.06	3.32	14.60
2230-BG	16	0.12	6.583	28.35	68.8	1,737400	264130	153000	43400	33400	21850	12480	57.90	16.44	12.64	8.28	4.74	39.80	11.30	8.70	5.69	3.26	13.90
2230-BC	12	0.06	6.550	21.05	65.2	1,217942	186010	101700	34000	25100	16700	8510	54.70	18.30	13.60	9.00	4.50	35.60	11.92	8.80	5.86	2.93	15.00
2230-BB	12	0.12	5.300	26.30	65.0	1,228080	231950	127500	38200	32350	20700	13200	55.00	16.50	13.85	8.93	5.72	35.80	10.73	9.00	5.80	3.72	16.00
SECTION II. ANTHRACITE—STOVE SIZE																							
2230-BK	16	0.03	12.000	17.60	72.9	2,098472	174750	104800	31100	21100	11440	6310	60.00	17.80	12.10	6.55	3.55	43.75	12.97	8.82	4.78	2.59	8.80
2230-BI	16	0.06	9.833	20.80	72.5	2,020307	205300	121000	35700	25300	15000	8300	59.00	17.36	12.30	7.31	4.03	42.80	12.59	8.91	5.30	2.92	10.20
2230-BM	16	0.09	7.417	26.40	66.7	1,775576	239450	140500	38430	30900	18540	11030	58.62	16.08	12.93	7.74	4.63	39.10	10.70	8.63	5.16	3.09	11.00
2230-BL	16	0.12	7.083	29.10	71.7	2,012595	284470	166000	44400	37400	23700	12970	58.40	15.60	13.14	8.32	4.54	41.90	11.20	9.43	5.97	3.25	13.00
2230-BJ	12	0.06	6.167	25.00	64.5	1,354230	219510	124400	39100	28800	17700	9510	56.70	17.80	13.10	8.05	4.35	36.60	11.50	8.45	5.19	2.81	12.10
2230-BH	12	0.12	5.167	29.70	70.2	1,415545	274030	149500	46300	39600	25000	13630	54.60	16.90	14.45	9.10	4.95	38.30	11.85	10.14	6.38	3.47	15.50
SECTION III. ANTHRACITE—CHESTNUT SIZE																							
2230-I	16	0.03	17.500	12.57	71.3	2,127309	121550	78600	21100	12500	6770	2580	64.50	17.40	10.20	5.60	2.30	46.00	12.40	7.28	4.00	1.64	6.00
2230-P	16	0.06	11.200	19.57	69.0	2,007070	179100	104700	31350	23000	12950	7100	58.50	17.50	12.50	7.20	4.00	40.30	12.10	8.84	4.97	2.76	9.50
2230-R	16	0.09	9.300	23.50	65.7	1,936722	207650	118000	36300	28100	16450	8800	56.90	17.50	13.50	7.90	4.20	37.40	11.50	8.87	5.19	2.76	11.00
2230-Q	16	0.12	7.750	27.00	63.7	1,922694	230780	131300	40200	30800	19500	8980	56.90	17.40	13.90	8.40	4.40	36.23	11.10	8.85	5.35	2.16	12.70
2230-M	12	0.06	7.550	22.90	64.8	1,507080	199260	117000	34750	25450	15430	6850	58.70	17.40	12.75	7.74	3.41	38.00	11.30	8.27	5.02	2.21	10.40
2230-L	12	0.12	5.670	30.10	63.4	1,466224	258700	143000	44400	35200	24100	12000	55.30	17.10	13.60	9.25	4.75	35.10	10.85	8.62	5.87	3.01	12.10
2230-K	8	0.03	5.780	19.55	65.0	997089	172450	105100	31550	19700	11450	4650	61.00	18.30	11.40	6.60	2.70	39.70	11.90	7.41	4.29	1.75	8.40
2230-N	8	0.06	4.500	25.00	61.8	929308	206200	120000	36600	23400	17150	9320	58.00	17.70	11.30	8.30	4.70	35.90	10.94	6.98	5.13	2.91	11.40
2230-O	8	0.09	3.810	29.70	61.5	938420	245860	136400	43700	32860	20900	12000	55.50	17.80	13.40	8.50	4.80	34.10	10.95	8.24	5.23	2.95	12.80
2230-J	8	0.12	3.250	35.10	59.3	916576	282000	156700	55500	34200	23350	12300	55.70	19.70	12.20	8.30	4.10	33.10	11.70	7.24	4.92	2.43	14.10
SECTION IV. ANTHRACITE—PEA SIZE																							
2230-AN	10	0.03	10.500	13.88	67.6	1,268387	120730	73500	21606	14500	7680	3450	60.90	17.90	12.00	6.30	2.90	41.30	12.10	8.12	4.26	1.96	7.00
2230-AO	10	0.06	7.634	18.82	68.0	1,255086	164470	94400	27950	22600	12900	6820	57.40	17.00	13.75	7.85	4.00	39.00	11.56	9.35	5.34	2.72	8.50
2230-AP	10	0.09	6.750	22.65	65.0	1,278035	189320	105400	32700	26900	15800	8520	55.70	17.25	14.50	8.35	4.50	36.20	11.20	9.25	5.42	2.92	9.75
2230-AQ	10	0.12	5.833	26.20	64.0	1,256709	215400	112000	37600	33300	20200	12300	52.00	17.45	15.50	9.35	5.70	33.20	11.16	9.92	5.98	3.65	11.20
2230-AM	8	0.06	5.983	19.90	66.7	1,022167	170790	97000	29400	23800	13470	7120	56.80	17.20	13.90	8.50	4.20	37.85	11.48	9.27	5.27	2.80	9.30
2230-AL	6	0.06	3.670	25.80	60.5	737963	201040	112600	35200	27800	16200	9240	56.00	17.50	13.80	8.10	4.60	33.90	10.60	8.35	4.90	2.78	9.90
SECTION V. ANTHRACITE—BUCKWHEAT SIZE (INDUCED DRAFT)																							
2230-AV	10	0.06	12.500	10.80	63.7	1,153548	92190	54200	16720	12000	5050	3320	58.80	18.15	13.05	6.40	3.6	37.45	11.56	8.31	4.08	2.29	6.60
2230-AU	10	0.09	10.250	13.50	68.4	1,270337	123820	70100	21750	17700	9550	4730	56.55	17.55	14.50	7.70	3.8	38.7	12.0	9.85	5.27	2.46	8.00
2230-AT	10	0.12	8.330	16.22	66.5	1,204530	144670	80400	25650	21050	11930	5640	55.50	17.75	14.50	8.25	4.0	36.9	11.75	9.64	5.48	2.66	8.30
2230-AS	8	0.06	8.918	12.40	68.0	1,007853	113040	66700	20000	14500	7910	3930	59.00	17.68	12.84	7.09	3.48	40.15	12.00	8.74	4.76	2.37	6.75
2230-AR	6	0.06	6.000	14.87	67.1	801083	133506	73780	25250	18860	10500	5116	55.20	18.90	14.20	7.87	3.83	37.0	12.7	9.53	5.28	2.57	8.50

[a]Loss to dry flue gas.

[*Continued on next page*]

Table 3 (continued)—Results of All Tests Showing Conditions of Fuel Bed Depth, Draft, Output and Per Cent Output of Each Section

Test No.	Coal Feed Notches	Air Setting	Duration Hours	Coal Per Hour	Overall Eff.	Total Output Btu	Output, Btu Per Hour						Output, Sq Ft Steam Radiation						Output, Per Cent of Total				
							Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5
SECTION VI. ANTHRACITE—BUCKWHEAT—STOKER FIRED (WITHOUT FURNACE BAFFLE)																							
2230-BQ	4	4.50	8	7.65	78.0	636190	79580	30080	26820	11790	6670	4220	331.5	125.0	112.0	49.1	27.8	17.6	37.8	33.8	14.8	8.4	5.2
2230-BP	7	5.20	8	13.81	71.5	1,036510	129580	49895	40080	20480	11968	7160	539.5	207.5	167.0	85.4	49.8	29.8	38.5	30.95	15.8	9.25	5.5
2230-BO	10	6.35	8	19.60	66.2	1,383550	172940	65730	50570	28390	17790	10460	720.5	274.0	211.0	118.0	74.0	43.5	38.0	29.2	16.4	10.3	6.1
2230-BN	13	6.40	8	24.20	65.7	1,696920	212130	88200	54800	34150	21400	13580	883.1	367.5	228.0	142.0	89.1	56.5	41.5	25.8	16.1	10.1	6.5
SECTION VII. ANTHRACITE—BUCKWHEAT—STOKER FIRED (WITH 21 IN. BAFFLE 10 IN. ABOVE TUYERE)																							
2230-BR	10	6.35	8	18.62	65.6	1,302750	162820	88500	30300	21700	14000	8320	678.8	369.0	126.2	90.5	58.4	34.7	54.4	18.6	13.3	8.6	5.1
2230-BS	13	6.40	8	23.90	68.0	1,732260	216480	122080	39460	28150	18200	8590	900.0	508.0	164.0	117.3	74.9	35.8	56.4	18.2	13.0	8.42	3.98

Test No.	Fuel Bed Depth Inches	Draft	Duration Hours	Coal Per Hour	Overall Eff.	Total Output Btu	Output, Btu Per Hour						Output, Sq Ft Steam Radiation						Output, Per Cent of Total				
							Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5
SECTION VIII. BY-PRODUCT COKE																							
2230-Z	16	0.03	6.000	22.80	66.4	1,137383	190840	106000	32500	25900	18620	7820	795.3	442.0	135.0	108.0	77.7	32.6	55.8	17.1	13.2	9.8	4.1
2230-AA	16	0.06	4.800	28.10	61.2	1,071534	223320	120800	38600	31400	20800	11720	930.8	503.0	161.0	131.0	86.8	49.0	54.1	17.3	14.1	9.3	5.2
2230-AB	16	0.09	3.883	34.40	59.5	1,033544	265990	146200	40600	37700	26650	14840	1105.8	610.0	169.0	157.0	111.0	61.8	55.0	15.25	14.15	10.0	5.6
2230-AC	16	0.12	3.467	38.20	61.0	1,049527	302859	166000	46500	44300	28050	18000	1262.0	692.0	193.5	184.5	117.0	75.0	54.9	15.3	14.6	9.3	5.9
2230-AH	12	0.06	3.506	30.00	60.5	825074	236000	131000	39500	33200	21850	10460	982.5	545.0	164.4	138.0	91.5	43.0	55.5	16.7	14.1	9.3	4.4
2230-AG	8	0.06	2.250	33.30	55.4	539318	239700	134000	41300	33600	20600	10200	998.0	558.0	172.0	140.0	85.6	42.4	55.9	17.2	14.0	8.6	4.3
SECTION IX. PETROLEUM COKE																							
2230-U	16	0.03	4.750	24.10	56.5	1,017403	214000	130000	30400	26400	16880	10320	892.0	542.0	126.6	110.0	70.3	43.1	60.8	14.2	12.3	7.9	4.8
2230-T	16	0.06	4.000	27.70	55.0	961233	240310	140500	34900	31350	20070	13490	1001.0	585.0	145.8	130.6	83.6	56.0	58.5	14.5	13.0	8.4	5.6
2230-S	16	0.09	3.734	30.90	54.6	994110	266070	151500	40800	33400	23000	17370	1108.2	631.0	170.0	139.0	95.8	72.4	56.9	15.3	12.6	8.7	6.5
2230-V	16	0.12	3.300	33.70	53.4	935665	283560	167000	38600	35600	26460	15900	1182.0	696.0	161.0	148.4	110.3	66.3	58.9	13.6	12.6	9.3	5.6
2230-W	12	0.06	3.150	25.62	54.0	687667	218230	125000	34030	28700	18500	12000	909.4	521.0	141.8	119.5	77.1	50.0	57.5	15.6	13.1	8.5	5.4
2230-X	8	0.06	2.300	21.40	52.5	408692	177650	95900	28650	25000	17250	10850	740.4	400.0	119.3	104.0	71.9	45.2	54.0	16.1	14.1	9.7	6.1

Test No.	Percent Cont.[b]	Duration Hours	Lb Oil Per Hour	Total Output Btu	Output, Btu Per Hour						Output. Sq Ft Steam Radiation						Output, Per Cent of Total				
					Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5
SECTION X. OIL BURNER WITHOUT BAFFLE																					
2230-BW	25	8	3.16	364885	45595	23500	9990	6950	3360	1795	190.0	98.0	41.5	29.0	14.0	7.5	51.54	21.9	15.25	7.37	3.94
2230-BV	50	8	6.34	788930	98615	51005	21755	15195	8060	2600	411.5	212.2	90.6	63.3	34.6	10.8	51.8	22.0	15.4	8.16	2.64
2230-BU	75	8	9.47	1,208175	151025	77930	32990	22995	11420	5690	629.2	324.5	137.5	95.9	47.6	23.7	51.63	21.82	15.22	7.56	3.77
2230-BT	100	8	12.53	1,592862	199110	105530	42485	29010	14940	7145	825.1	440.0	172.0	121.0	62.3	29.8	53.0	21.3	14.58	7.5	3.62
SECTION XI. OIL BURNER WITH BAFFLE																					
2230-BX	50	8	6.25	788750	98560	59250	18500	11700	6360	2750	410.8	247	77.0	48.8	26.5	11.5	60.2	18.75	11.82	6.45	2.78
2230-BY	100	8	11.78	1,493177	186540	112820	37270	20360	11130	5100	777.7	470	155.3	84.8	46.4	21.2	60.4	20.0	10.9	5.96	2.74

Test No.	Percent Cont.[b]	Duration Hours	Cu Ft Gas Per Hour	Total Output Btu	Output, Btu Per Hour						Output, Sq Ft Steam Radiation						Output, Per Cent of Total				
					Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Total	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5	Sec. 1	Sec. 2	Sec. 3	Sec. 4	Sec. 5
SECTION XII. GAS BURNER																					
2230-CC	25	8	107.5	318193	39745	25900	7400	3680	1845	920	165.7	108	30.8	15.4	7.7	3.8	65.2	18.6	9.25	4.64	2.31
2230-CB	50	8	200.0	630085	78764	49700	15050	8300	4340	1374	328.1	207	62.7	34.6	18.1	5.7	63.1	19.1	10.53	5.51	1.76
2230-CA	75	8	294.4	943047	117850	75300	21850	11970	6550	2180	491.2	314	91.0	49.8	27.3	9.1	63.9	18.53	10.15	5.55	1.87
2230-BZ	100	8	382.5	1,258233	157300	101000	28450	15750	8700	3400	655.5	421	118.5	65.6	36.2	14.2	64.2	18.1	10.0	5.54	2.16

[b] Per cent of time in operation.

E. E. Ashley Takes Partners

E. E. Ashley has announced that H. H. Bond and C. L. Chase have been taken into partnership under the firm name of Edward E. Ashley, consulting engineer. The practice of mechanical, electrical and sanitary engineering will be continued with offices at 10 East 40th St., New York, N. Y.

E. H. Beling in Contracting Business

E. H. Beling has entered the contracting field in heating, ventilating and air conditioning, specializing in school work. Mr. Beling was formerly manager of the unit ventilator division of The Herman Nelson Corp. and his new business will be conducted under the name of The Beling Engineering Co., at 2428 13th St., Moline, Ill.

A.I.E.E. Elects New Officers

During the annual summer meeting of the *American Institute of Electrical Engineers,* at Pasadena, Calif., the election of the following officers to serve for the year beginning August 1, 1936, was announced: *President*—A. M. MacCutcheon, Cleveland, O.; *Vice-Presidents*—A. C. Stevens, Schenectady, N. Y., O. B. Blackwell, New York, N. Y., C. F. Harding, Lafayette, Ind., L. T. Blaisdell, Dallas, Tex., C. E. Rogers, Seattle Wash.; *Directors*—K. B. McEachron, Pittsfield, Mass., C. A. Powel, East Pittsburgh, Pa., R. W. Sorensen, Pasadena, Calif.; *Treasurer*—W. I. Slichter, New York, N. Y.

These officers together with the following hold-over officers will constitute the Board of Directors for the next administrative year: E. B. Meyer, Newark, N. J., J. B. Whitehead, Baltimore, Md., C. V. Christie, Montreal, Que., R. H. Fair, Omaha, Neb., Mark Eldredge, Memphis, Tenn., W. H. Harrison, Philadelphia, Pa., N. B. Hinson, Los Angeles, Calif., F. M. Farmer, New York, N. Y., N. E. Funk, Philadelphia, Pa., H. B. Gear, Chicago, Ill., C. R. Jones, New York, N. Y., P. B. Juhnke, Chicago, Ill., W. B. Kouwenhoven, Baltimore, Md., E. S. Lee, Schenectady, N. Y., L. W. W. Morrow, New York, N. Y., and G. C. Shaad, Lawrence, Kan.

Andre Merle Associates Organized

New offices have been established by André Merle Associates, consultants, at 11 East 44th St., New York, N. Y., specializing in the drafting of plans and specifications, engineering, surveys, installation and operation cost analysis of air conditioning, automatic heating, mechanical ventilation and commercial refrigeration installations.

Contractors Association Meets in Philadelphia

Members of the *Heating, Piping and Air Conditioning Contractors National Association* assembled at the Bellevue-Stratford, Philadelphia, Pa., May 25-27, for the 47th annual convention of the association and elected the following officers: *President*—J. H. Zink, Baltimore, Md.; *Vice-President*—G. H. Dickerson, Chicago, Ill.; and *Treasurer*—M. J. Chagnard, Cleveland, O. Five members were elected to the Board of Directors: J. M. Anderson, Philadelphia, Pa.; Thomas O'Callaghan, Boston, Mass.; E. W. Tompkins, Albany, N. Y.; S. A. Pope, Chicago, Ill., and G. J. Stoehr, Milwaukee, Wis.

H. G. Black, president of the *H. P. & A. C. C. Philadelphia Association,* welcomed the members and among the subjects of discussion at the sessions were: Report of Committee on Trade Promotion, J. H. Zink, *chairman;* a Clinic on the Sales Problems of the Industry when A. R. Herske presented the Manufacturer's Viewpoint, W. A. Brecht the Wholesaler's Viewpoint and J. L. DeNeille and J. C. Fitts the Contractor's Viewpoint, and S. L. Land spoke on How to Sell Air Conditioning with Radiator Heat; Voluntary Codes by J. P. Baldwin; The Government in Industry, by C. M. Proctor; Report of Committee on Air Conditioning by H. E. Wetzell, *chairman;* the Plumbing and Heating Industries Bureau by G. H. Dickerson; Air Conditioning from a Promotional Angle by W. L. Fleisher; Report of Committee on Certification by T. F. Hanley, Jr., *chairman;* Report of Boiler Output Committee by H. M. Hart, *chairman;* Housing Problem in the U. S. by F. S. Fitzpatrick; Cooperation with the Construction League and Chamber of Commerce of the U. S. by J. H. Zink, and Social Security Legislation by V. M. Miles.

Bert Bernstrom Air Conditioning Consultant

Bert Bernstrom has opened offices at 844 Rush St., Chicago, Ill., heading an organization to give complete service as air conditioning consultant for manufacturers, architects, contractors and prospective buyers of air conditioning systems and equipment.

Design Problems of Heating and Ventilation

The fundamental engineering problems which are continually being met in the fields of Heating, Ventilating and Air Conditioning have been considered in many text books, but A. T. Henly, of Mottingham, Kent, England, seeks to avoid repetition of such matter as far as possible in Design Problems of Heating and Ventilation.

It is the author's aim to present special problems and many of the methods of calculation or applications of known theories are definitely unique and the graphical representation of formulae and graphical methods of calculations are of great importance. With this in view the author has eliminated, wherever possible, tables evaluating known formulae and substituted instead graphical methods of representation.

Included in this book are detailed chapters covering the following subjects: hot water heating; hot and cold water supply systems; ventilation; industrial air conditioning; drying systems; district heating and hot water supply methods; steam distribution in boiler plants; dust and fume extraction; steam removal and pneumatic conveying; swimming baths and public washhouses; and special requirements of factories and schools.

This book is available from the publisher, Crosby Lockwood & Son Ltd., Amen Corner, E. C. 4, London, England.

Westinghouse Inspection Trip

On June 17, 1936, several engineers and editorial writers were invited to participate in a preview of the Westinghouse hermetically sealed condensing unit at the factory in East Springfield, Mass.

After observing several of the new engineering features of the direct drive complete assembled unit, the group was escorted through the newly arranged plant for manufacturing the equipment. A. J. Offner, Consulting Engineer of New York City, was an invited guest and acted as the official representative of the Society.

At luncheon in the Hotel Kimball, Springfield, Mass.

Local Chapter Reports

A. S. H. V. E. Chapter Organized at Montreal

May 11, 1936. The organization meeting of the Montreal Chapter of the A.S.H.V.E. was held at the Queen's Hotel. Dinner was served at 6:30 p. m. and the business session was called to order at 7:45 by G. L. Wiggs, who presided. Sixty members and guests were present.

The first order of business was the appointment of Mr. Wiggs as temporary chairman and C. W. Johnson as acting secretary.

A message of welcome to all those attending was presented by Mr. Wiggs, who also explained the purpose of the meeting. He then read the petition for a Montreal Chapter and gave a brief outline of the contemplated program for meetings during the coming season.

The principal speaker was E. H. Gurney, Toronto, 2nd Vice-President of the Society. In his talk, Mr. Gurney discussed activities of the A.S.H.V.E. and the benefits of membership, as well as the advantages of participating in local chapter work and attendance at Annual and Semi-Annual Meetings of the Society.

A. V. Hutchinson, secretary, was the second speaker and presented a talk on the History of the A.S.H.V.E., outlining the relationship between each chapter and the national organization.

G. H. Osborne, A. W. Givin and Leo Garneau were appointed to the Nominating Committee.

A discussion was held regarding a regular meeting night and resulted in the selection of the second Monday of each month for a dinner meeting.

It was moved, seconded, and unanimously carried that the proposed $3.00 annual local chapter dues be accepted.

The report of the Nominating Committee presented the following slate of officers, who were elected upon motion for a single cast of ballot:

President—G. L. Wiggs.
Vice-President—F. J. Friedman
Secretary—C. W. Johnson.
Treasurer—F. G. Phipps.
Board of Governors—A. B. Darling T. E. McGrail, J. P. Linton.

President Wiggs presented the need for a committee to draft the Constitution and By-Laws for the Chapter and F. R. Barnsley, J. R. Robertson and G. B. Elliott were elected.

The following were nominated, seconded and elected as members of the Membership Committee: A. Lamontagne, W. J. Pead, H. Gendron, J. Bews, J. J. Fraser, M. T. Dixon, S. S. Pitt, F. R. Barnsley, T. Worthington and C. W. Johnson.

The proposal of having a preliminary meeting in June was discussed and it was suggested that the possibility of such a meeting should be investigated by the Board of Governors.

Before adjournment formal announcement of the date of the Charter Meeting was given as October 13, when the principal speaker will be Prof. G. L. Larson, Madison, Wis., president of the A.S.H.V.E.

Southern California Chapter Hears Talk on Foreign Trade

May 12, 1936. The regular meeting of the Southern California Chapter took place at the Rosslyn Hotel, Los Angeles, and dinner was served at 7:00 p. m. with 15 members and guests present.

A. H. Simonds presided in the absence of Pres. Leo Hungerford.

Secy. H. M. Hendrickson read correspondence from headquarters office regarding the Semi-Annual Meeting of the Society at Buck Hill Falls, Pa., and also relating to regulations and dues for Student Members.

The speaker of the evening was Pedro Cezon, president, Export Corp., Ltd., who gave a talk in an interesting manner on Latin American Relations in Connection with Foreign Trade. One of the local Chapter members, M. I. Blumenthal, then made a few remarks concerning his experiences with heating, ventilating and refrigeration in South America.

Mr. Simonds presented the revised draft of the proposed air conditioning code for Southern California and Arizona and copies were distributed to the members for study and discussion at the next meeting.

Measuring Instruments and Development of Air Conditioning Discussed by Massachusetts Chapter

May 11, 1936. Forty-five members and guests of Massachusetts Chapter attended dinner and the May meeting at the Engineers Club, Boston.

Pres. W. A. McPherson announced the election of W. T. Jones, C. P. Yaglou and John Turner to serve as members of the Board of Governors for a three-year period.

Professor Yaglou then explained the operation of two measuring instruments developed at the Harvard School of Public Health to aid in the study of atmospheric conditions and comfort.

An instrument called a heated globe thermometer was designed to indicate the temperature next to the body of a normally clothed person, compensating for the factors of air temperature, air motion and temperature of the environment. Three concentric spheres are arranged at approximately the same temperature gradient as the inner and outer layers of clothing, with a thermometer inserted in the inner sphere to indicate temperature next to the body. The outer surface absorbs heat at approximately 97 per cent efficiency corresponding to dark clothing. In order to simulate a sweaty body condition, part of the globe is covered with a wet cloth. With the aid of this special thermometer formulae may be derived from the known comfort temperature range, 83-87 next to the body, to determine required air temperature, air velocity or environment temperature as in the case of panel heating.

The second instrument described was called a heated thermometer anemometer, used for measuring low air velocities where the usual anemometer is inaccurate. The bulb of an ordinary glass tube thermometer is wrapped with resistance wire, which is connected to a source of current, usually a battery, and the temperature noted when fully heated. The change in temperature caused by air motion is recorded by the swinging arm method and a chart is developed. Professor Yaglou stated that he believed the instrument was almost too sensitive for ordinary practice as it will indicate a variation of 0.1 ft at velocities from 0—50 fpm with an accuracy of ½ per cent in this range and 1 per cent to 4000 fpm.

Philip Drinker, chairman of Papers and Meetings Committee, then introduced J. L. Levine, air conditioning department, General Electric Co., who discussed the Developments of Air Conditioning from the viewpoint of a manufacturer of such equipment. Slides were shown of testing laboratories where equipment is subjected to operation under varying conditions, and where facilities are provided for testing, heating and cooling equipment, as well as for studying air motion, heat loss through walls and sound volume.

Mr. Levine spoke of the simplification of engineering methods in calculating heat loss and gain, citing a building in Boston which required 8 hours to calculate 3 years ago. By means

the practice which has been developed, an equally accurate result can now be obtained in 5 min.

Looking forward to the future, Mr. Levine discussed methods of heating now being studied, including panel heating and the use of off peak electric heating using the storage tank method.

After a question and discussion period, Secy. G. B. Gerrish reports that the meeting closed with thanks to both speakers for their interesting presentations.

May 22, 1936. At a meeting of the Board of Governors, the following officers were elected to serve for the season of 1936-37:

President—James Holt
Vice-President—G. B. Gerrish
Secretary—John Turner
Treasurer—Edgar Shaw.

President Holt and W. A. McPherson, past president, were elected to serve as councillors to the *Engineering Societies of New England.*

A motion was passed giving the president authority to change the day of meeting to the second Tuesday of each month, if deemed desirable.

Social Meeting and Election of Officers for Manitoba Chapter

May 28, 1936. Fifteen members of Manitoba Chapter and their wives and friends met at the Fort Garry Hotel, Winnipeg, and after dinner, a bridge party was enjoyed.

Mrs. J. B. Steele presented awards to Miss Bowden, ladies' first, William Glass, gentlemen's first, and D. F. Michie, consolation.

At a short business session, officers were elected for the ensuing year as follows:

President—J. B. Steele.
Vice-President—D. F. Michie.
Secretary—C. H. Turland.
Treasurer—H. R. Eade.
Board of Governors—Frank Thompson, William Glass, J. E. Yates.

Wisconsin Chapter Elects Officers, Honors Life Member, and Views Pictures on Control Systems

May 25, 1936. Wisconsin Chapter held its final business meeting of the season at the City Club, Milwaukee. A. S. Krenz and I. J. Haus were appointed as tellers of election, and on hearing their report, the Chapter voted to cast a unanimous ballot for the following officers:

President—M. E. Erickson.
Vice-President—W. H. Wilson.
Secretary—R. G. Koch.
Treasurer—L. A. Burch.
Board of Governors—G. L. Larson, Ernest Szekely, C. H. Randolph.

Annual reports from the president and treasurer were read which were encouraging. C. H. Randolph, in his report as president, discussed the growth and strength of the Chapter, showing that it has doubled its membership in the last 3 years even though inactive members have been eliminated from the roster. He also disclosed that 36 of the 40 members have paid their dues for 1936. The report of W. H. Wilson as treasurer was equally optimistic in that the largest cash balance in the history of the Chapter was recorded.

The meeting was turned over to President Erickson, who gave a short inaugural address.

Prof. G. L. Larson, president of the A.S.H.V.E., was then called upon to present the certificate of Life Membership in the Society to H. W. Ellis. Mr. Ellis accepted the honor with a delightful talk on the duties and responsibilities of a member. He modestly claimed that he had never done anything outstanding for the Society, but, that as an army is dependent on the individual soldiers, so the Society is also dependent for success on its individual members, and he hoped that he had been a good soldier.

Two talking pictures on control systems for heating, ventilating and air conditioning installations were presented by Charles Cochran. Secretary Koch has reported that the films were interesting and instructive and were enjoyed by the members.

Golf Tournament Enjoyed by Michigan Chapter Members, Who Elect New Officers

May 20, 1936. The annual golf tournament and meeting of Michigan Chapter was held at Meadowbrook Country Club. Approximately 40 members played golf in the afternoon and the dinner and meeting was attended by 51 members and guests. In the absence of Pres. A. C. Wallich, Vice-Pres. R. K. Milward presided.

The first order of business was the roll call of members present, after which Secy. W. F. Arnoldy read correspondence about the Semi-Annual Meeting of the Society at Buck Hill Falls, Pa., and the rate of dues for Student Members.

A detailed treasurer's report was given by F. J. Feely and the chairman of the Membership Committee called attention to the elimination of limited chapter membership by action of the Council.

Mr. Feely spoke of the difficulty of some members in meeting dues in arrears and presented a motion for consideration of the Council.

G. D. Winans took charge of the meeting and the slate of the Nominating Committee for Chapter officers for the coming year was presented. After nominations for each office had been closed, it was voted that the secretary be instructed to cast a unanimous ballot for the following officers:

President—R. K. Milward.
Vice-President—F. J. Feely.
Secretary—G. H. Tuttle.
Treasurer—A. P. Darlington.

The names of members presented by the Nominating Committee for the Board of Governors, of whom three were to be selected, were read and the tellers reported the election of R. F. Connell, Thomas Chester and F. R. Bishop as members of the Board of Governors.

The new officers were then installed and introduced to the meeting.

J. F. McIntire paid tribute to W. F. Arnoldy, for his work as secretary during the past two seasons.

President Milward introduced S. H. Downs, Kalamazoo, Mich., former president of Western Michigan Chapter and at present a member of the Council of the Society.

The meeting was turned over to the chairman of the Golf Committee, E. J. Anderson, who with the assistance of H. E. Paetz, distributed the following awards: E. E. Dubry for attendance, H. M. Armstrong for low gross, and E. J. Anderson for second low gross which was tied with A. P. Darlington. The award for the greatest number of par holes was tied between E. D. Sheeley and H. E. Paetz and was awarded to Mr. Sheeley. A. P. Darlington had the most birdies and H. E. Paetz the least number of putts. The award for high gross score went to Edward Glanz, while F. J. Feely and Bill Olds tied for the greatest number of putts, with Mr. Feely being awarded the trophy.

Kickers prizes went to Messrs. R. K. Milward, W. G. Boales, R. F. Connell, R. L. Spitzley, J. F. McIntire, A. E. Fisher, F. S. Lorne, G. Pfeiffer, W. F. Arnoldy, J. C. McFate, F. Marzolf, G. D. Winans, G. H. Tuttle, M. Shea, J. Lanze, and F. R. Bishop.

President Milward appointed Messrs. Arnoldy and Boales as a committee to audit the books of the treasurer for 1935-36 and the meeting adjourned at 10:30 p. m.

Western Michigan Chapter Holds Annual Meeting and Elects Officers

May 25, 1936. The fifth annual meeting of the Western Michigan Chapter was held at the Greenridge Country Club and was well attended by 40 members and guests.

Entertainment and play were the features of the day with golf and horseshoes prevailing. The Chapter golf cup went to P. O. Wierenga to be held until 1937. Awards were made to Walter Pearl for blind bogey, to Tom Osberger for second blind bogey, and to Emil Bauch for the best dressed golfer. The horseshoe pitching contest was won by Prof. L. G. Miller.

Dinner was served at 7:00 p. m. and, following presentation of the awards, a Liars Contest was featured. John Wieland scored

easily with his tale of servicing a hot water heating system and finding it a spawning bed for smelt.

The business session was called to order by President Wierenga and an Auditing Committee consisting of W. W. Bradfield and S. H. Downs approved the treasurer's report.

The tellers for the election of officers for the season of 1936-37 announced the following officers:

President—L. G. Miller.
Vice-President—J. H. Van Alsburg.
Secretary—E. I. Adams.
Treasurer—C. R. McConner.
Board of Governors—S. H. Downs, P. O. Wierenga, S. W. Todd.

The meeting was then turned over to the president, Professor Miller, and it was moved and supported that a special meeting be held June 16 to entertain A.S.H.V.E. members attending the meeting of the *National District Heating Association* in Grand Rapids.

Mr. Wierenga was presented with an engraved cigarette case in appreciation of his work as president during the past year and attention was called to his previous service as secretary for two terms.

President Miller declared the meeting closed at 11:00 p. m. with everyone present feeling that the annual party had again been a success.

J. J. Troske reports that the Chapter looks forward to a successful season next year, knowing that their officers are particularly capable and enthusiastic.

Residential Air Conditioning Discussed at Open Meeting of Cleveland Chapter

April 14, 1936. One hundred members and guests attended an open meeting at the Cleveland Club, which was preceded by a dinner and business meeting when 25 members were present.

A letter from A. V. Hutchinson, secretary, was read, suggesting that the chapter discuss constructive suggestions for the improvement of THE GUIDE and that such recommendations be forwarded to the Guide Publication Committee.

It was brought to the attention of the meeting that limited chapter membership had been discontinued by action of the Council.

The following members were elected to the Chapter's Nominating Committee to report at the May meeting: C. F. Eveleth, *chairman*, A. L. Vanderhoof, and G. B. Longcoy.

Membership applications were received and approved from H. S. Ackley, J. W. Booker and T. J. Jordan, and it was decided to publish a chapter membership list after the election of officers.

Before adjournment of the business meeting at 8:30, Pres. G. L. Tuve urged the members to study THE GUIDE 1936 so that it could be properly discussed at the next meeting.

C. A. Thompson, building contractor, was the first speaker at the open meeting on Residential Air Conditioning. He explained that the general contractor is still a listener in regard to air conditioning and that his ideas on the subject are based almost entirely on the information given him by engineers.

Mr. Thompson stated that to the home owner residential air conditioning is largely winter air conditioning, while summer air conditioning is at present remote, experimental and expensive.

E. P. Farris discussed the subject of The Distribution of Air and cited the procedure followed by the Central Bureau for Heating and Air Conditioning. He emphasized the importance of proper air motion in the home and stated that ducts are designed on the equal friction loss basis with 300-350 fpm velocity at the register face, 500 fpm in risers, and 1000 fpm in main trunks.

Mr. Farris recommended that supply grilles be placed low on the inside walls and that the air temperature at the register be 150 deg with 0 deg outside. He also recommended 80 deg room temperature for bathrooms and warned of drafts due to the air hitting the opposite wall.

The present forms of winter air conditioning were described in a paper by F. R. Dickenson, outlining the functions that complete winter air conditioning equipment should perform, namely, air cleaning, air heating, humidification, introduction of fresh air, and automatic control. He also discussed humidity control and pointed out that present installations of winter air conditioning cannot be readily adapted to summer air conditioning.

Mr. Dickenson then outlined the requirements for summer air conditioning to be as previously given, with the exception that the air must be cooled instead of heated, and dehumidified instead of humidified. He mentioned the lack of authentic engineering data and standard equipment for residential cooling as compared to residential heating.

C. E. Hansen spoke of the importance of economy in first cost, operation and maintenance of large residential air conditioning systems, and emphasized that designs should be simple and fully automatic.

Mr. Hansen recommended that supply registers be placed in the floor under the windows to prevent high ceiling temperatures and stated that in his opinion it was not practical to cool kitchens and bathrooms because of the variable cooling load. The speaker recommended filtering both return air and fresh air and that special caution be taken to prevent leakage in the duct system.

Secy. Philip Cohen has reported that this interesting meeting adjourned at 11:00 p. m., after a question and answer period.

Election of Officers and Entertainment Feature Western New York Meeting

May 11, 1936. Western New York Chapter held its May meeting at King Arthur's Restaurant, Buffalo, and Pres. W. E. Voisinet welcomed the 62 members and guests who attended.

After reading minutes of the previous meeting and communications, President Voisinet called attention to the Society's Semi-Annual Meeting at Buck Hill Falls, Pa., June 22-24.

Left to Right (Back Row): C. A. Gifford, L. P. Saunders, and J. J. Landers. (Front Row): B. C. Candee and P. S. Hedley

The following officers were elected to serve during the coming season:

President—P. S. Hedley.
1st Vice-President—B. C. Candee.
2nd Vice-President—J. J. Landers.
Secretary—C. A. Gifford.
Treasurer—L. P. Saunders.

Mr. Voisinet then presented the gavel to the newly elected president, Mr. Hedley. The other officers were introduced and President Hedley gave a short outline of his plans for the coming year.

Joseph Davis, chairman of the Entertainment Committee, was then presented and took charge as master of ceremonies. After a full hour of real entertainment, refreshments were served and a rousing vote of thanks was given to Mr. Davis for his efforts in arranging so successful a party.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 30 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by July 15, 1936, these candidates will be ballotted upon by the council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ALLEN, W. W., Mgr. & Engr., American Coolair Corp., Jacksonville, Fla.	J. L. Varner W. M. Marsh (*Non-Member*)	R. E. Lynch (*Non-Member*) J. L. Elder (*Non-Member*)
BALDI, Giuseppe, Engr., Compagnia Italiana Westinghouse, Torino, Italy	Enrico Donzelli Aldo Gini	Giuseppe Manzoli (*Non-Member*) Luigi Greppi (*Non-Member*)
BELING, E. H., Owner & Chief Engr., The Beling Engrg. Co., Moline, Ill. (*Advancement*)	G. E. Otis H. W. Nelson	R. H. Nelson H. A. Beighel
BENSINGER, MARK, Sales Engr., Combustioneer Stoker Corp., Washington, D. C.	J. B. Hewett A. E. Beitzell	G. S. Frankel D. M. Robinson
BYRD, TOM, Sales, American Rolling Mill Co., Middletown, Ohio	J. H. Van Alsburg A. W. Williams	F. L. Myers A. V. Hutchinson
COREY, G. R., Vice Pres., Appliance Engineers, Hartford, Conn.	Homer Addams J. A. Darts	R. S. Graham (*Non-Member*) Herbert Malley (*Non-Member*)
DYKES, J. B., Estimator, T. A. Morrison & Co., Ltd., Montreal, Que., Can.	C. W. Johnson A. W. Givin	G. H. Osborne G. L. Wiggs
EDWARDS, J. D., Asst. Director of Research, Aluminum Co. of America, New Kensington, Pa.	F. C. Houghten W. A. Danielson	Percy Nicholls J. T. Kingsley (*Non-Member*)
FLINN, G. S., Chief Engr., McGregor's, Inc., Memphis, Tenn.	A. R. MacMillan C. E. Lewis	H. H. Hill M. J. Stevenson
FRASER, J. J., Dist. Mgr., Minneapolis-Honeywell Reg. Co., Montreal, Que., Can.	G. L. Wiggs T. E. McGrail	F. G. Phipps Thomas McDonald
GITTLESON, HAROLD, Sales Mgr., Lariviere, Inc., Montreal, Que., Can.	E. H. Gurney J. P. Linton	G. L. Wiggs A. W. Givin
HARDING, E. R., Br. Mgr., Kewanee Boiler Corp., Greensboro, N. C.	R. B. Dickson W. H. Sullivan (*Non-Member*)	Harry Barton (*Non-Member*) J. M. Hunt (*Non-Member*)
HUFF, J. M., Mgr. Air Cond. Dept., Silkensen & Co., Inc., Galveston, Tex.	A. R. MacMillan J. A. Kiesling	H. H. Hill W. B. Cochran
KNAPP, D. S., Br. Mgr., Chamberlin Metal Weather Strip Co., Inc., Minneapolis, Minn.	A. J. Huch A. B. Algren	G. A. Dahlstrom R. C. Jordan
KROEKER, J. D., Pres., Columbia Engrg. Co., Portland, Oregon	E. C. Willey G. D. Kingsland	D. R. Munro (*Non-Member*) S. H. Grof (*A.S.M.E.*)
LENIHAN, W. O., Owner, W. O. Lenihan, Inc., Buffalo, N. Y.	J. J. Landers B. C. Candee	P. S. Hedley Joseph Davis
LOCKHEAD, K. Y., Supt. of Bldg's. Office, Hudson's Bay Co., Winnipeg, Man., Can.	C. H. Turland D. F. Michie	William Glass A. H. Doe (*Non-Member*)
MAIER, A. H., Supervisor of Htg. & Vtg., Board of Education, Middletown, Ohio	K. A. Wright W. C. Green	G. B. Houliston T. D. Boyd (*Non-Member*)
MYTINGER, K. L., Mgr., Air Cond. Div., Fitzgibbons Boiler Co., New York, N. Y.	Homer Addams G. E. Olsen	J. A. Darts H. J. Rose
NELSON, E. L., Engrg. Dept., The Union Ice Co., Los Angeles, Calif.	E. H. Kendall A. H. Simonds	W. E. Barnum, Jr. Harrold English
PAGE, V. C., Director of Sales, Automatic Htg. & Cooling Systems, Inc., Baltimore, Md.	Donald Luty F. C. McIntosh	H. H. Erickson Wm. Sanford (*Non-Member*)

CANDIDATES	*Proposers*	*Seconders*
PETERSON, C. M. F., Instructor in Mech. Engrg., Mass. Inst. of Tech., Cambridge, Mass.	H. C. Moore James Holt	W. H. Jones (*A.S.R.E.*) G. B. Wilkes (*Non-Member*)
PREECE, L. W., Sales Mgr. & Engr., Electric Specialty Co., Erie, Pa.	C. H. Graham V. P. John	H. B. Joyce B. C. Candee
ROWE, W. M., Sales Engr., American Blower Corp., Cleveland, Ohio	H. M. Nobis Philip Cohen	G. L. Tuve F. R. Dickenson
SCHWARTZ, S. B., Consulting Engr., New Orleans, La.	G. E. May O. E. Gammill, Jr.	C. C. Nifong (*Non-Member*) W. J. Amos (*Non-Member*)
SELIG, E. T., JR., Treas., Fuel Savers, Inc., Harrisburg, Pa.	W. A. Danielson I. H. Geiger	H. C. Eicher Homer Addams
SOLSTAD, L. L., Student, University of Minnesota, St. Paul, Minn.	F. B. Rowley A. B. Algren	J. V. Martenis B. J. Robertson (*Non-Member*)
STACY, L. D., Sales, Ilg Elec. Vtg. Co., Chicago, Ill.	M. K. Arenberg J. F. Hale	H. J. Lagodzinski B. L. Casey
WAGNER, E. A., Pres., Wagner Engrg. Corp., Pittsfield, Mass.	Elliott Harrington P. S. Lyon	D. W. McLenegan L. Crandall (*Non-Member*)
YOUNG, J. T., Jr., Sales Engr., Crane Co., Salt Lake City, Utah.	C. V. Haynes F. H. Gaylord	O. E. Davis L. B. Hampton (*Non-Member*)

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and ballotted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ALLEN, A. W., Sales Engr., Pease Foundry Co., Ltd., Toronto, Ont., Can.

BERGHOEFER, V. A., Vice Pres., Sterling Engrg. Co., Milwaukee, Wis. (*Advancement*)

CARTER, J. H., Special Repr., Frick Co., St. Louis, Mo.

DICKENSON, F. R., Dist. Mgr., American Blower Corp., Cleveland, Ohio (*Advancement*)

ERICKSON, E. V., Engrg. Dept., Carrier Engrg. Corp., Newark, N. J.

HALE, F. J., Mgr. & Partner, Empire Sheet Metal Works, Ltd., Vancouver, B. C.

HUGHES, W. U., Managing Director, The Lewis-Brown Co., Ltd., Montreal, Que., Can.

KRENZ, A. S., Sales Mgr., Krenz & Co., Milwaukee, Wis. (*Advancement*)

MCLEAN, J. E., Supt. Bldgs. & Grounds Dept., University of Pittsburgh, Pittsburgh, Pa.

MOULD, D. E., Mgr., J. W. Mould & Son Plbg. & Htg., Edmonton, Alta., Can.

POLING, D. B., Engr., Lamneck Prod., Inc., Columbus, Ohio

RODEE, E. J., Engr., John B. Pierce Lab. of Hygiene, New Haven, Conn.

TOLHURST, G. C., Htg. Expert, The Gurney Fdry. Co., Ltd., Montreal, Que., Can.

WIGGS, G. L., Consulting Engr., Montreal, Que., Can. (*Advancement*)

ASSOCIATES

ALLONIER, H. R., Dist. Sales Mgr., Buckeye Blower & J. J. Nesbitt Co., Columbus, Ohio

BALLANTYNE, G. L., Mgr., Htg. Sales Dept., Crane, Ltd., Montreal, Que., Can.

BENOIST, R. E., Mgr., Benoist Bros. Supply Co., Mt. Vernon, Ill.

DAY, I. M., Mfrs. Repr., Washington, D. C.

DUFAULT, F. H., Htg. Expert, General Steel Wares, Ltd., Montreal, Que., Can.

FRANCE, C. N., Mgr., Colonial Fuel Oil, Inc., Washington, D. C.

FRAZIER, J. E., Secy. Treas., Simplex Engrg. Co., Washington, D. C.

FRIEDMAN, ARTHUR, Pres., Air Controls, Inc., Cleveland, Ohio

HARTLINE, W. R., Vice Pres. & Treas., Combustioneer Stoker Corp., Washington, D. C.

HUNT, MCDONALD, Mfrs. Repr., Baltimore, Md.

LAMONTAGNE, A. F., Sales Mgr., The Gurney Fdry. Co., Ltd., Montreal, Que., Can.

MARCONETT, V. G., Engr. & Factory Supt., The Farquhar Furnace Co., Wilmington, Ohio

SAPP, C. L., Sales Mgr., The Farquhar Furnace Co., Wilmington, Ohio

SHILLING, H. C., Sales Engr., Barber-Colman Co., Chicago, Ill.

VOLBERDING, L. A., Chief Air Cond. Engr., Norge Corp., Detroit, Mich.

VOYE, V. J., Sales, Richardson & Boynton Co., Boston, Mass.

WARD, J. J., Pres. & Mgr., Wenzler & Ward, Inc., Seattle, Wash. (*Reinstatement*)

WOOLLEY, J. H., Vice Pres., Woolley Coal Co., Inc., Maplewood, N. J.

JUNIORS

BORKAT, PHILIP, Test Engr., Sales Dept., Air Controls, Inc., Cleveland, Ohio

DAVIDSON, J. C., Gas Htg. Inspector, City of Minneapolis, Minneapolis, Minn.

GRIEST, KERMIT, Sheet Metal Worker, Frank Limbach Co., Pittsburgh, Pa.

JOSEPHSON, SIMON, Plbg. & Htg. Engr., Astor Plbg. & Htg. Corp., Brooklyn, N. Y.

KIMMELL, P. M., Engr., L. R. Krumm Co., Columbus, Ohio

MARTIN, LEONARD, Sales Engr., H. L. Peiler & Co., Montreal, Que., Can.

PARK, H. E., Sales, Shaw-Perkins Mfg. Co., Pittsburgh, Pa.

SCOTT, L. G., Asst. to Supt. of Bldgs., Hudson's Bay Co., Winnipeg, Man., Can.

SOUTHMAYD, R. T., Sales Engr., American Blower Corp., Cleveland, Ohio.

YATES, J. E., Jr., Service & Installation Mgr., Automatic Htg. Co., Ltd., Winnipeg, Man., Can.

STUDENTS

LEDGETT, F. D., Student, University of Toronto, Toronto, Ont., Can.

MCMAHON, J. E., Student, University of Ill., Urbana, Ill.

TURNER, H. S., Jr., Student, Agric. & Mech. College of Texas, College Station, Texas

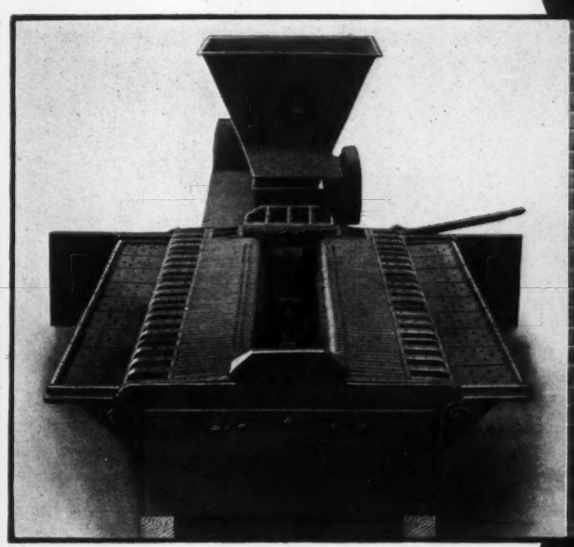
DESIGN

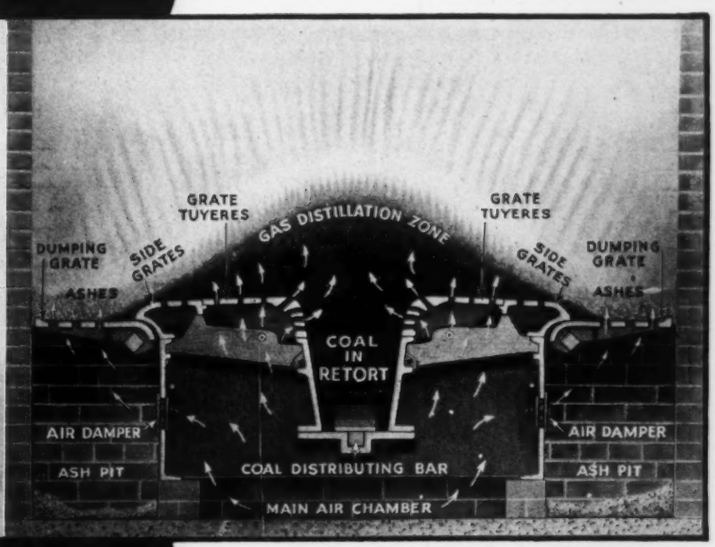
SIMPLE AND ACCESSIBLE yet "has everything" necessary for efficient, dependable results because of: ADJUSTABLE PLUNGER FEED • SIDE CLEANING • MECHANICAL DRIVE • Detroit LoStokers combine the same high standards of design and workmanship as the larger power Detroit Stokers.

Built for Hard Service

CONSTRUCTION

COMPLETELY BUILT in our own Works. All interior parts are made of special alloy, high heat resisting iron. The heavy drive consists of machine cut worms and gears mounted in Timken Bearings, runs in oil and requires little power for operation.





APPLICATION

CAREFULLY STUDIED from an engineering standpoint. Detroit LoStokers are built in many sizes. We have just the size to fit your furnace and provide sufficient grate area to efficiently handle your heavy load and yet operate economically with light loads.

PERFORMANCE

EFFICIENT because the coal is pushed into the retort and as it rises, just the proper amount of air is supplied for complete combustion. No special coal is necessary. Respond to sudden changes of load and maintain uniform steam pressure because they are automatically controlled.

DETROIT LoSTOKERS

DETROIT STOKER COMPANY
Sales and Engineering Offices:
Fifth Floor, General Motors Bldg., Detroit, Mich.
Works at Monroe, Mich. . . . District Offices in Principal Cities
BUILT IN CANADA AT LONDON, ONTARIO

Write for Bulletin 378

MODERNIZE AND ECONOMIZE WITH

DETROIT SINCE 1898 STOKERS

Code for Ventilation and Air Conditioning

Following is Part XIII of the heating code of Minneapolis, Minn., dated 1936, covering ventilation and air conditioning, continued from last month.

1304.2. The difference in temperature between the entering air and the room air for horizontally discharging supply openings in side walls of rooms with conventional registers when cooling shall never exceed 2 degrees per foot in height from the floor to the bottom of the opening.

Horizontally discharging air inlets in rooms when placed higher than head level shall be designed for sufficient velocity of discharge to bring about diffusion without drafts, especially when the entering air is cooler than the room air. The minimum design air inlet velocity shall be 100 linear feet per minute. The entering air velocity may be rapid for such openings, provided that high speed jets of air do not rebound from columns, walls, etc., so as to create discomfort to people.

1304.3. Upward discharging air inlets in rooms shall be designed with arrangements, especially with delivering air cooler than the air in the room, so that a velocity of at least 300 linear feet per minute may be obtained so as to promote diffusion.

Section 1305. **Refrigerating Apparatus and Refrigerants:** All refrigerants, cooling equipment, piping and specialties installed and used in connection with air conditioning shall be subject to the approval of the Department of Buildings.

Section 1306. **Duct Design and Construction, Fans, Air Washers, Air Cleaning Devices, etc.:** In all air conditioning installations the metals used shall be of moisture resisting character and of such weight and with such bracing as will prevent vibration. The ducts shall be substantially air tight.

Section 1203 and its Sub-Sections, Sections 1207, 1208 and 1209 of this ordinance together with the standards incorporated in Chapters 14 to 22, inclusive, Volume 13, *American Society of Heating and Ventilating Engineers Guide*, 1935 Edition, shall govern the design and construction of all duct systems, fans, blowers, air washers, air cleaning devices, cooling coils and other devices in connection with all air conditioning systems.

Section 1307. **Humidifying and Filtering Equipment:** All convectors where condensation might occur when cooling shall have drain pans with tubing running to drip with an air break above a trapped sewer connection or its equivalent.

1307.1. No fixed spray humidifier or spray dehumidifier shall be installed above the basement of a building unless there shall have been installed under it a water tight safing graded to a drain, so that in case of leakage or overflow there shall be no danger of water damage to the rooms under the apparatus.

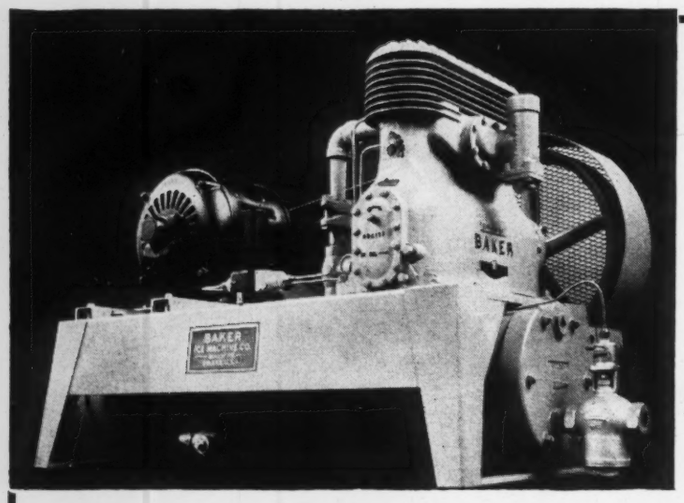
Section 1308. **Noise Control:** All moving apparatus used in air conditioning shall be fastened securely to its own foundations, which should be a single unit for both driver and driven apparatus, and this foundation shall be carried on a noise-inert substance such as rubber in shear, or cork under and around the edges of the foundation, without any direct contact between the foundation and the building structure.

1308.1. No metal duct shall connect directly to any fan or housing without a sound-inert junction, such as of canvas.

1308.2. Piping connections from pumps where practicable shall be made with rubber hose, with the machinery on a floating foundation.

1308.3. Sharp edges of metal facing an air current shall be rounded or streamlined rather than left sharp and rough. Partitions, housings, and large ducts must be so heavy and so well braced that there will be no vibrating or rattling when air is being delivered.

1308.4. On account of noise danger the following air velocities shall not be exceeded in those parts of the ducts



BAKER

UNITS FOR AIR CONDITIONING SERVICE

especially designed for ***FREON REFRIGERANT***

Baker Automatic Units were among the first machines to successfully handle Freon Refrigerant without leakage of gas through shaft seal. Baker pioneered the metal shaft seal for small and medium capacity compressors and has perfected its efficiency during the past three years of heavy demand for compact units capable of high-capacity service. Engineers and contractors are asked to compare the performance of Baker Freon Units with other makes for efficiency, design, and cooling power. Full specifications and descriptive matter on Baker Units sent on request.

MEMBER RELIABILITY ACMA EXPERIENCE AIR CONDITIONING MANUFACTURERS ASSOCIATION

BAKER ICE MACHINE COMPANY, INC.

1509 Evans, Omaha, Nebr.

Sales and Service in All Principal Cities.

How to Absorb NOISE in Air Ducts

Three Burgess Bulletins

● Less than 20 minutes reading time will provide you with all the information and data required to understand the nature of noise, how to determine the amount of duct lining for any air conditioning system, and how to install Burgess Acousti-Pad noise absorption duct lining.

Every sheet metal contractor, air conditioning engineer, and architect should have these important data sheets in his reference file. All essential information on quieting duct noise is thus available in convenient form. No need to delve into obscure physics books, no need for logarithms, algebra, or calculus. Charts and tables given in these three basic reference sheets are extremely valuable in solving acoustic problems connected with air ducts quickly and efficiently.

Burgess Acousti-Pad, recommended for duct lining, is a patented application of a perforated facing over a sound absorbing material. It is manufactured by the Acoustic Division of the Burgess Battery Company, licensed under the C. F. Burgess Laboratories, Inc., patents.

with Burgess Acousti-Pad

● The sound absorbent faced with perforated hard surfaced material called Burgess Acousti-Pad has both acoustic and mechanical advantages over baffles and other methods of reducing duct noise. The perforated pattern in Burgess Acousti-Pad is designed to be most effective on those noises to which our ears are most sensitive. The hard perforated surface offers no more restriction to air flow than smooth metal duct. Air velocities as high as 3000 feet per minute will not harm this modern duct lining.

Other advantages of Burgess Acousti-Pad are: odorless, flameproof, germ-proof, does not mold or rot, provides thermal insulation, permanent—requires no servicing, simple to install, and reasonably priced. (Only a small portion of most duct systems needs to be lined.)

Complete information is contained in three Burgess Bulletins.

These three valuable bulletins are FREE!

Paste this coupon to your letterhead or to a postcard now!

BURGESS ACOUSTIC

MFRS. OF AIR CLEANERS, INTAKE SILENCERS, MUFFLERS, ENGINE HOODS, COMMUNICATION BOOTHS, AND ACOUSTIC DUCT LINING

● Burgess Battery Co., Dept. H
111 W. Monroe St., Chicago, Ill.

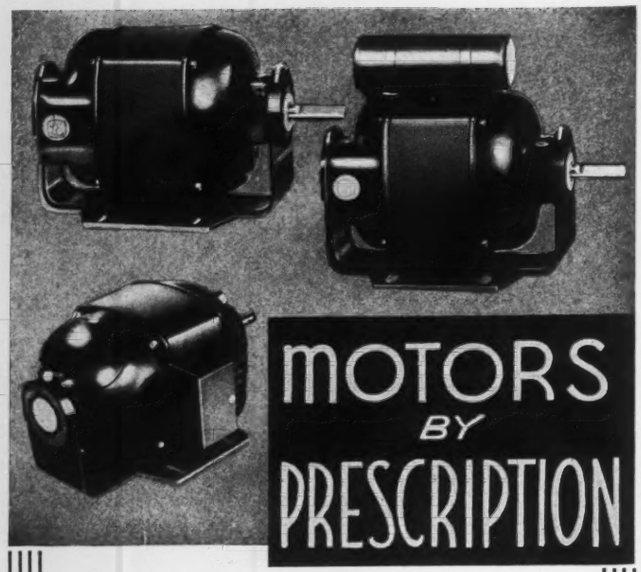
Please send immediately the three Burgess Bulletins on How to Absorb Noise in Air Ducts.

Firm Name.....

Address.....

City..... State.....

Your Name.....



Your requirement may be a motor to drive an oil burner, a propeller-fan, a unit heater or an air conditioner, in which cases the Leland split phase motor, especially designed for such appliances, is indicated.

Or you may be in search of the most effective motor available for stoker operation which suggests the Leland repulsion-induction design.

And so it goes—"motors by prescription"—motors with particular characteristics for particular applications — motors that are special purpose rather than general purpose designs — also specials where required.

The Leland line is a complete one. State your requirement. Get a Leland and test it.

THE LELAND ELECTRIC CO.
Dayton, Ohio

Canadian Address Toronto
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LELAND
DAYTON . . . TORONTO . . . PARIS . . . MILAN
MOTORS

of air conditioning systems closer than about 50 feet from the room outlets.

Fan outlet 1,500; Trunk duct 1,200; Through free area of convectors 1,000 linear feet per minute.

1308.5. Where fans, motors, compressors, and the like are placed in rooms under or otherwise adjacent to occupied rooms, provision shall always be made to reduce noise transmission. Suggested means are double walls, double ceilings, sound absorbent material and the like.

Section 1309. **Insulation:** Insulation on pipes and air-ways when heating, is designed to reduce transmission to the surrounding air, and is under no danger of becoming moist except in case of a liquid leak.

1309.1. Insulation on pipes and air-ways when cooling, in addition to reducing heat transfer from the surrounding air to the cooler substance inside, must prevent the surrounding air from reaching its dewpoint and thus depositing moisture. The dewpoint temperatures for conditions of the air around a duct or pipe may be read from a psychrometric chart.

1309.2. In air conditioning in which refrigeration is used, all chilled surfaces must be insulated not only as a measure of control of heat waste, but also to prevent damage due to water drip.

1309.3. For ducts carrying air not colder than 40 F, a ½ inch layer of board or quilt insulation will serve, provided that the chilled areas are all covered so that air does not pass against the cold surface. For housing around evaporators and for spray dehumidifier housings ½ inch of quilt insulation and ½ inch of board insulation with alternating joints and thorough sealing, have given satisfactory service. Cork 1 inch thick properly sealed, gives satisfactory results, especially for spray dehumidifier housings which carry direct expansion evaporators also.

1309.4. For low side refrigerant piping cork insulation is acceptable, of thickness commensurate with the temperature difference.

1309.5. For systems which use cold water piped to the convectors for summer only, any anti sweat pipe insulation 1 inch thick, is acceptable. Where the same pipes convey hot water in winter and cold water in summer, a combination insulation shall be used, which will withstand both extremes. Compressed mineral wool has this faculty, though there is evidence that any good quality steam pipe insulation will serve satisfactorily provided that there are no exposed chilled surfaces to condense the damaging dew.

1309.6. **Covering and Lining Incombustible, When:** In all buildings of Type 1 construction and in all Class C and D buildings, as defined in the Minneapolis Building Code as adopted March 29, 1934,[2] all ducts therein shall be lined or covered with incombustible material only, where lining or covering is required. Pipe covering installed in buildings of the above type and class shall be incombustible.

[2]The classification of buildings as defined in the Minneapolis Building Code, adopted March 29, 1934, relates to the occupancy, and the types relate to the construction of a building.

"Type 1 construction" covers fireproof or fire-resistive construction, including walls, enclosures, vertical members, partitions, and all that is included in the general construction of a building.

"Class C Buildings" are those buildings occupied as a public or semi-public or similar types of buildings, such as armories, auditoriums, churches, club houses, court houses, dance halls, drill halls, gymnasiums, libraries, lodge halls, museums, post offices, railway passenger stations, schools, seminaries, and skating rinks.

"Class D Buildings" include all theaters and all buildings provided with an auditorium and stage provided with fly galleries for movable scenery, and all buildings containing an auditorium used frequently for viewing moving pictures, dramatic or theatrical performances to which admission is customarily charged either in money or other consideration.

A. C. M. A. Air Conditioning Application Standards

Application engineering standards for comfort air conditioning have been adopted by the *Air Conditioning Manufacturers' Association*, 915 Southern Bldg., Washington, D. C. These standards, released June 12, are actually "recommended practice" for *A.C.M.A.* members, "pending the formulation of an adequate national standard for air conditioning application under the sponsorship of the proper technical societies." It is expected that

AUDITORIUM PATENTS

COVERING CONTRIBUTIONS TO THE AIR CONDITIONING ART

WILL BE ENFORCED

There need be no misunderstanding of the effect of the recent decision of the Second Circuit Court of Appeals holding invalid certain claims of the Lewis Reissue Patent No. 16,611 and of the Fleisher Patent No. 1,670,656.

The judgment in that suit is personal to the defendants there involved;

it does not invalidate the remaining claims of those patents, two of which have been held valid and infringed in other litigation.

It does not involve or invalidate the

TWENTY-FIVE ADDITIONAL PATENTS

and numerous applications owned or controlled by Auditorium covering other important improvements and economies in the field of air conditioning.

Auditorium has already brought suit against another infringer and expects to enforce its rights wherever evidence of unlicensed use of its inventions is obtained.

Permission to operate under Auditorium's patents may be secured from any of the following Licensees, all of whom have acknowledged the validity of the Auditorium patents and agreed to pay the royalties thereunder.

AMERICAN BLOWER CORPORATION
Detroit, Michigan

CARRIER ENGINEERING CORPORATION
850 Frelinghuysen Avenue, Newark, N. J.

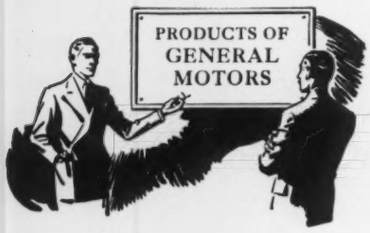
GENERAL ELECTRIC COMPANY
Schenectady, New York

ROSS INDUSTRIES CORPORATION
New Brunswick, N. J.

THE COOLING & AIR CONDITIONING CORP.
Hyde Park, Boston, Mass.

YORK ICE MACHINERY CORPORATION
York, Pa.

AUDITORIUM CONDITIONING CORPORATION
New York Office—17 EAST 42nd STREET



A name

that builds confidence

•

Delco-Heat offers automatic equipment for every type of heating system . . . all are products of General Motors

THERE is a great expansion taking place in the automatic heating business. Thousands of homes are about to install some form of oil or gas heat.

Delco-Heat is ready with a full line of equipment ranging from moderate size conversion burners to complete heating and air conditioning plants.

During a few short years Delco-Heat has moved from scratch to a position among the front line leaders in automatic heating. This tremendous growth in sales has been due to superior products backed by a name that builds confidence.

Like all other products of General Motors, Delco-Heat automatic heating equipment is built to insure owner satisfaction. Products of General Motors find ready acceptance, and give the service that purchasers expect.

DELCO-FRIGIDAIRE CONDITIONING CORPORATION

AUTOMATIC HEATING **DAYTON, OHIO** *AIR CONDITIONING*

PRODUCTS OF GENERAL MOTORS

there will be some further minor modifications during the ne month or so in these standards.

Personnel of the *A.C.M.A.* technical and code committee r sponsible for the formulation of the standards is: Donald Frenc Chairman, Carrier Engineering Corp.; F. H. Faust, Gener Electric Co.; Elliott Harrington, General Electric Co.; A. Karlson, Parks-Cramer Co.; C. E. Lewis, Delco-Frigidaire Co ditioning Corp.; A. J. Mallinckrodt, Baker Ice Machine Co., Inc.; L. S. Morse, York Ice Machinery Corp.; L. A. Philipp Kelvinator Corp.; R. E. Robillard, Frigidaire Corp.; Harr Sloan, The Vilter Mfg. Co.; R. M. Stikeleather, B. F. Sturte vant Co.; E. R. Wolfert, Westinghouse Electric & Mfg. Co.; J. R. Zwickl, De La Vergne Engine Co.

A. Scope and Purpose—This code is limited to applica tion engineering standards and practices for determining the con ditions and the loads for which to design systems for condi-

Table 1—Outside Air Conditions for Heating Estimates

The following table lists the recommended outdoor winter temperature to use as the basis for preparing heating estimates. These figures should be used unless the customer specifies the temperature upon which the estimate shall be made.

The figures given in this table are not the extreme minimum temperatures recorded in each locality and it is to be expected that lower temperatures will be encountered. However, very low temperatures usually exist only for short periods. It is thus unnecessary and uneconomical to design the heating system for the extreme minimum temperature.

STATE AND CITY	DESIGN TEMP., F
ALABAMA	
Anniston	5
Birmingham	5
Gadsden	0
Mobile	15
Montgomery	10
Tuscaloosa	5
ARIZONA	
Bisbee	15
Flagstaff	-15
Globe	20
Nogales	20
Phoenix	25
Tuscon	20
Yuma	20
ARKANSAS	
Fort Smith	10
Hot Springs	5
Little Rock	5
Pine Bluff	10
Texarkana	10
CALIFORNIA	
Bakersfield	15
Fresno	25
Los Angeles	30
Montague	0
Pasadena	25
Sacramento	25
San Diego	35
San Francisco	35
San Jose	20
COLORADO	
Boulder	-15
Colorado Springs	-25
Denver	-20
Fort Collins	-30
Grand Junction	-15
Pueblo	-25
CONNECTICUT	
Bridgeport	-5
Hartford	-5
New Haven	-5
New London	-5
Norwalk	-5
Torrington	-10
Waterbury	-15
DELAWARE	
Dover	5
Milford	5
Wilmington	5
DIS. OF COLUMBIA	
Washington	0
FLORIDA	
Fort Myers	35
Gainesville	20
Jacksonville	25
Key West	45
Miami	35
Orlando	35
Pensacola	20
Tallahassee	25
Tampa	35
GEORGIA	
Athens	10
Atlanta	10
Augusta	15
Columbus	10
Macon	15
Rome	10
Savannah	15
Way Cross	20
IDAHO	
Boise	0
Lewiston	5
Pocatello	-5
Twin Falls	-10
ILLINOIS	
Aurora	-10
Bloomington	-10
Chicago	-10
Danville	-5
Decatur	-10
Elgin	-10
Joliet	-10
Peoria	-10
Rockford	-10
Springfield	-10
INDIANA	
Elkhart	-5
Evansville	0
Fort Wayne	-10
Indianapolis	-5
South Bend	-5
Terre Haute	-5
IOWA	
Burlington	-10
Cedar Rapids	-5
Clinton	-10
Council Bluffs	-10
Davenport	-10
Des Moines	-15
Dubuque	-10
Fort Dodge	-20
Keokuk	-10
Marshalltown	-15
Sioux City	-20
Waterloo	-15

(Part 2 of table on page 104)

"TUBE-TURNS MADE IT EASY!"

HERE'S just one small job which perfectly illustrates *several* advantages of Tube-Turns★. In the first place, both the odd-angle 130° fitting and the 45° elbow were cut from *one* 180° Tube-Turn! This was easily possible only because Tube-Turns have uniform wall-thickness at all points .... The full cross-section and the easy sweeping radius of these Tube-Turns will permanently reduce pumping costs .... The neatness, compactness and light weight of this Tube-Turn job speak for themselves. May we send you *all* the facts about Tube-Turns? Address: Tube-Turns, *Incorporated*, Herald-Post Building, Louisville, Kentucky.

★ The word "Tube-Turn" is NOT a general name for welding fittings. Tube-Turns are THE seamless fittings having uniform wall-thickness at all points, true circular cross-sections, uniform radius of curvature and smooth inner walls. Made ONLY by Tube-Turns, Incorporated.

tioning air for the comfort of persons, for installation within the United States.

These are minimum standards and are to be construed as defining recommended practice, rather than as hindering progress, or as preventing the use of other standards where such are justified by the economies of a case and where departures below good practice minimums are made clear to the buyer.

B. Name—The name "Air Conditioner" or "Air Conditioning" shall not be applied to any air treating combination which is not designed to embody the minimum complement of functions as defined below in Paragraph C. If a system embodies less, it shall be called by a name which describes only the functions performed, and does not incorporate the words "Air Conditioning" or "Air Conditioner."

C. Definitions—

1. *Application engineering standards* for air conditioning are those relating to conditions and factors which form the basis of design load estimating and specifications of performance. These are distinguished from standards of apparatus design, of installation practices, of trade practices and of safety.

2. *The design load* is the capacity required of the apparatus to maintain specified conditions inside when specified extreme conditions of temperature and humidity obtain outside and when all sources of load are taken at the maximum that will occur coincidentally, during periods of outside extremes.

3. *Design inside conditions* are the dry and wet bulb temperatures (or relative humidity) specified to be maintained inside at the time of occurrence of the design load.

4. *Design outside conditions* are the dry and wet bulb temperatures (or relative humidity) specified for design load computation.

5. *Comfort air conditioning* provides ventilation, air circulation, air cleaning and maintains temperature and humidity for the comfort of people.*

6. *An air conditioning system* provides ventilation, air circulation, air cleaning and equipment for maintaining temperature and humidity within prescribed limits.*

7. *An air conditioner* is a specific combination consisting of means for ventilation, air circulation, air cleaning and heat transfer with control means for maintaining temperature and humidity within prescribed limits.*

8. *A cooling (summer) air conditioner* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning, cooling and dehumidifying with control means for maintaining room temperature and humidity within prescribed limits.*

9. *A heating (winter) air conditioner* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning, heating and humidifying with control means for maintaining room temperature and humidity within prescribed limits.*

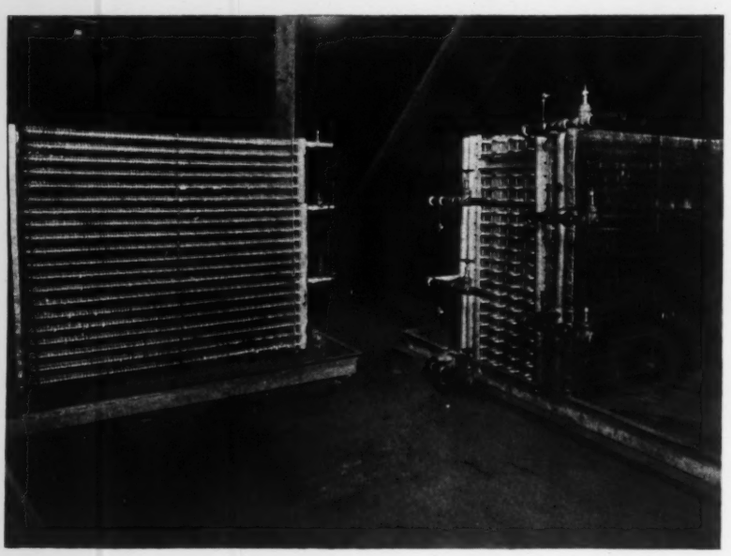
*Taken from *A.S.R.E.* Code for Rating and Testing Air Conditioning Equipment.

D. Design Load Factors—The following shall be specified as a basis for the calculation of design loads.

1. Design inside conditions.
2. Design outside conditions.
3. Number of occupants and other sources of substantial load from within doors.
4. Cfm per person assumed for ventilation.
5. Time of day at which maximum load is estimated to occur.
6. Class of activity assumed for occupants.

Calculations of design loads shall include the following sources of heat loss and heat gain:

1. *Heat loss* (winter load)
   a. Conduction through physical barriers, such as walls, doors, windows, floors, ceilings, etc.
   b. Heat and moisture required for incoming outdoor air.
2. *Heat gain* (summer load)
   a. Conduction through physical barriers, such as walls, doors, windows, floors, ceilings, etc.
   b. Heat from sunshine. (1) Direct effect through glass



BUSH COILS

installed with a brine system in a modern brewery. These coils have a 3¼" square fin spaced ¾" apart on 1" O. D. steel tube and are galvanized. However, there are 36 other standard combinations of fin and tube sizes and fin spacings available to meet all requirements and they can be made in all standard metals.

THE BUSH MFG. CO.

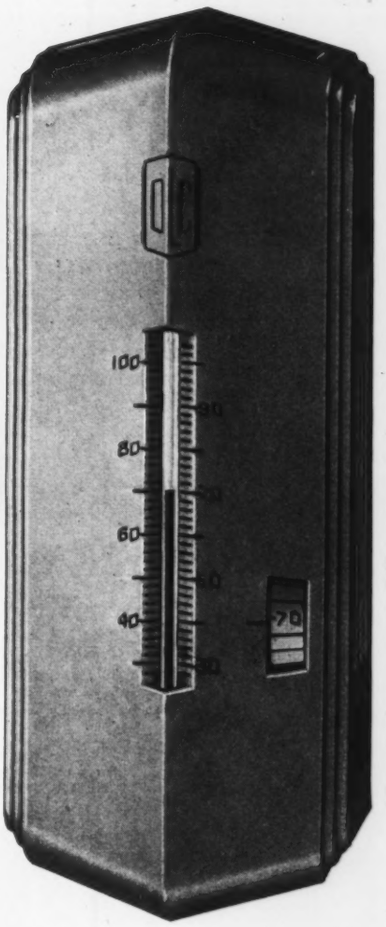
HARTFORD CONN.

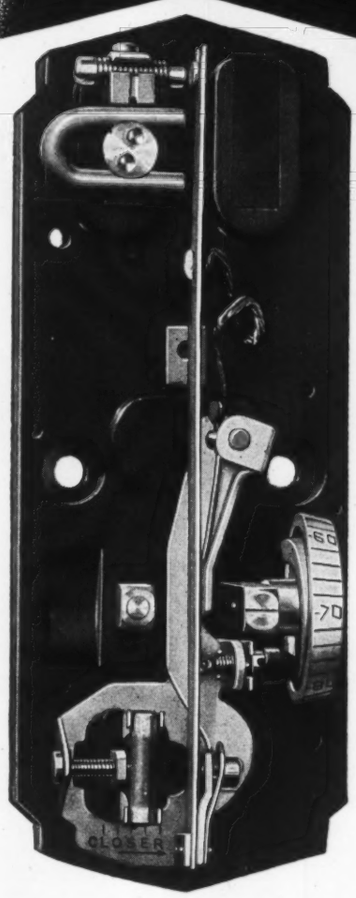
NEW YORK 489 FIFTH AVE.

DETROIT 6432 CASS AVE.

CHICAGO 610 N. OAKLEY BLVD.

WHY the two-eleven HEAT COMPENSATED THERMOSTAT IS DIFFERENT





Compensating type — cover removed to show adjustable heater element

THE Two-Eleven Thermostat with adjustable heat compensation operates differently from the conventional heat anticipating thermostat.

The heater element draws its current by a separate circuit—not through the "load." Thus one thermostat can be used for any load, without changing parts or replacing the instrument. The trade can stock *one thermostat* for use with *all* "Genuine Detroit" Controls or others on a circuit of 20-25 volts.

The heater element is easily adjustable after installation to meet the requirements of each job, regardless of exposure, type of construction or kind of heating plant.

The heater is scientifically designed and located so that, after adjustment on the job, extreme cold weather has practically no effect on the operating point.

Field adjustment of the Two-Eleven is not a critical operation. For the average installation this thermostat will give satisfactory regulation just as it is received from the factory.

The adjustable heater makes it unnecessary to change the "air gap" adjustment at the contacts. This is set at the factory to assure best response to heater effect.

The exterior is beautifully styled and finished to harmonize with modern interiors—designed to blend into its surroundings yet dignified and attractive in itself.

The Two-Eleven is also supplied without the heat compensating feature if desired. Both types in all models for heating, cooling, night and day, summer and winter and three wire.

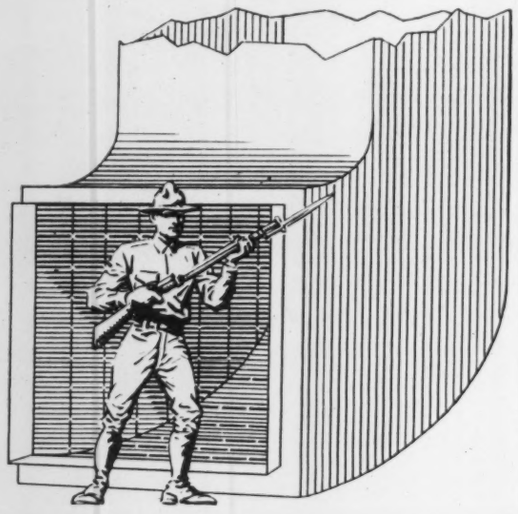
Write for details.

DETROIT LUBRICATOR COMPANY DETROIT, MICH. U. S. A.
5900 Trumbull Avenue

NEW YORK, N. Y.—40 West 40th St. • CHICAGO, ILL.—816 S. Michigan Ave.
LOS ANGELES, CALIF.—320 Crocker Blvd.

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

Canadian Representative—RAILWAY AND ENGINEERING SPECIALTIES LIMITED, Montreal, Toronto, Winnipeg



NOISE *Cannot Enter*
Air Ducts Guarded by
Ozite!

If you want to keep your air conditioning systems free from troublesome noises and vibrations . . . line the entrance of the air ducts with Ozite Hair Felt Duct Lining!

This resilient material, made entirely of sterilized and washed cattle hair, is unsurpassed as a sound deadener and has an especially high coefficient of sound absorption at the frequencies at which most motor and blower noises occur. It can be cut easily and applied quickly to any installation. Write for specifications and complete details now!

AMERICAN HAIR and FELT CO.
MERCHANDISE MART CHICAGO, ILL.

Ozite HAIR FELT
AIR DUCT LINING
and DUCT INSULATION
FOR ALL SOUND, HEAT AND
COLD INSULATION PURPOSES

areas exposed to the sun. (2) Additional conduction through opaque barriers, such as walls, roofs, etc., exposed to the sun.

c. Heat and moisture introduced by incoming outdoor air.

d. Heat and moisture liberated by occupants.

e. Heat and moisture liberated by appliances, illumination, combustion, etc.

The basis of all these design load values shall conform with the data in the current issue of the *A.S.H.V.E. Guide* except where these standards give other specific requirements.

E. Design Outside Conditions

1. *Heating*

a. The design outside dry bulb temperature for calculating heating load shall be that minimum temperature which has been recorded on more than 2 per cent of the days when heating is required. The local weather bureau information for the past 10 years shall be used as the authority for this record. For the purpose of this code, it is assumed that heating is required on any or all days when the average daily temperature is 65 F or less.

For convenience, Table 1 of this code gives approved temperatures to be used when local data is not available.

Table 1—Continued

STATE AND CITY	DESIGN TEMP., F
KANSAS	
Atchison	-10
Concordia	-10
Dodge City	-10
Leavenworth	-10
Salina	-15
Topeka	-10
Wichita	- 5
KENTUCKY	
Bowling Green	0
Frankfort	0
Hopkinsville	0
Lexington	0
Louisville	0
Owensboro	0
LOUISIANA	
Alexandra	20
Baton Rouge	20
New Orleans	25
Shreveport	15
MAINE	
Bar Harbor	-15
Belfast	- 5
Eastport	-10
Lewiston	-15
Millinocket	-15
Orono	-20
Portland	-10
Rumford	-20
MASSACHUSETTS	
Amherst	-10
Boston	0
Clinton	-10
Concord	-15
Fall River	-10
Fitchburg	-10
Lawrence	-10
Lowell	-15
New Bedford	0
Pittsfield	-10
Plymouth	- 5
Springfield	-10
Worcester	-10
Framingham	-10
MARYLAND	
Annapolis	5
Baltimore	5
Cambridge	5
Frederick	-5
Frostburg	-5
Salisbury	10

STATE AND CITY	DESIGN TEMP., F
MICHIGAN	
Alpena	-10
Ann Arbor	- 5
Big Rapids	-15
Calumet	-15
Detroit	0
Flint	-10
Escanaba	-15
Grand Rapids	-10
Kalamazoo	- 5
Lansing	-10
Ludington	-10
Marquette	-10
Muskegon	0
Port Huron	- 5
Saginaw	-10
Sault Ste. Marie	-20
MINNESOTA	
Alexandria	-25
Duluth	-25
Minneapolis	-20
St. Cloud	-25
St. Paul	-20
MISSISSIPPI	
Biloxi	15
Columbus	15
Hattiesburg	15
Jackson	15
Meridan	10
Natchez	15
Vicksburg	15
MISSOURI	
Columbia	-10
Hannibal	-10
Kansas City	- 5
St. Louis	- 5
St. Joseph	-10
Springfield	-10
MONTANA	
Anaconda	-15
Billings	-25
Butte	-20
Great Falls	-20
Havre	-30
Helena	-25
Miles City	-35
Missoula	-20
NEBRASKA	
Grand Island	-20
Hastings	-25
Lincoln	-10
Norfolk	-15
North Platte	-20
Omaha	-10
York	-15

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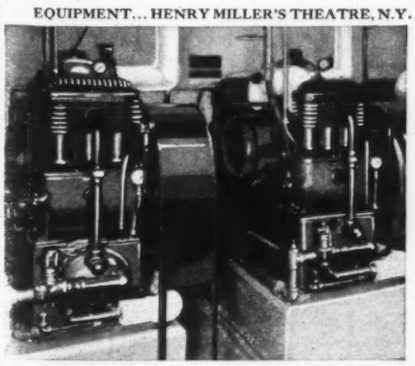

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Address_____

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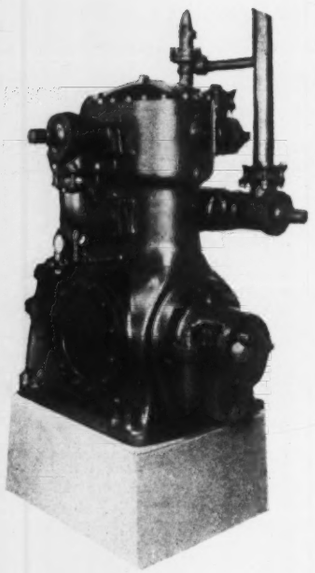

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2. *Cooling*

a. The design outside conditions for cooling shall be the average for ten years of the highest dry and wet bulb temperatures remaining each year after excluding those higher temperatures which have occurred on only ten days of each year and for durations of less than three hours.

Figs. 1 and 2 show maps of the United States, giving approximate design dry bulb and wet bulb temperatures which are approved for use until additional data is avail-

Fig. 1—Outside design dry bulb temperatures for cooling estimates

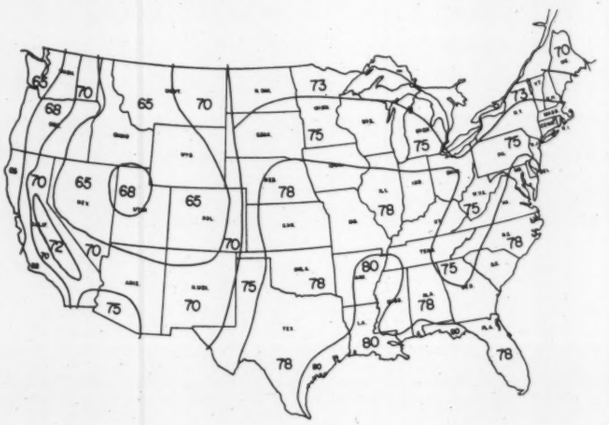


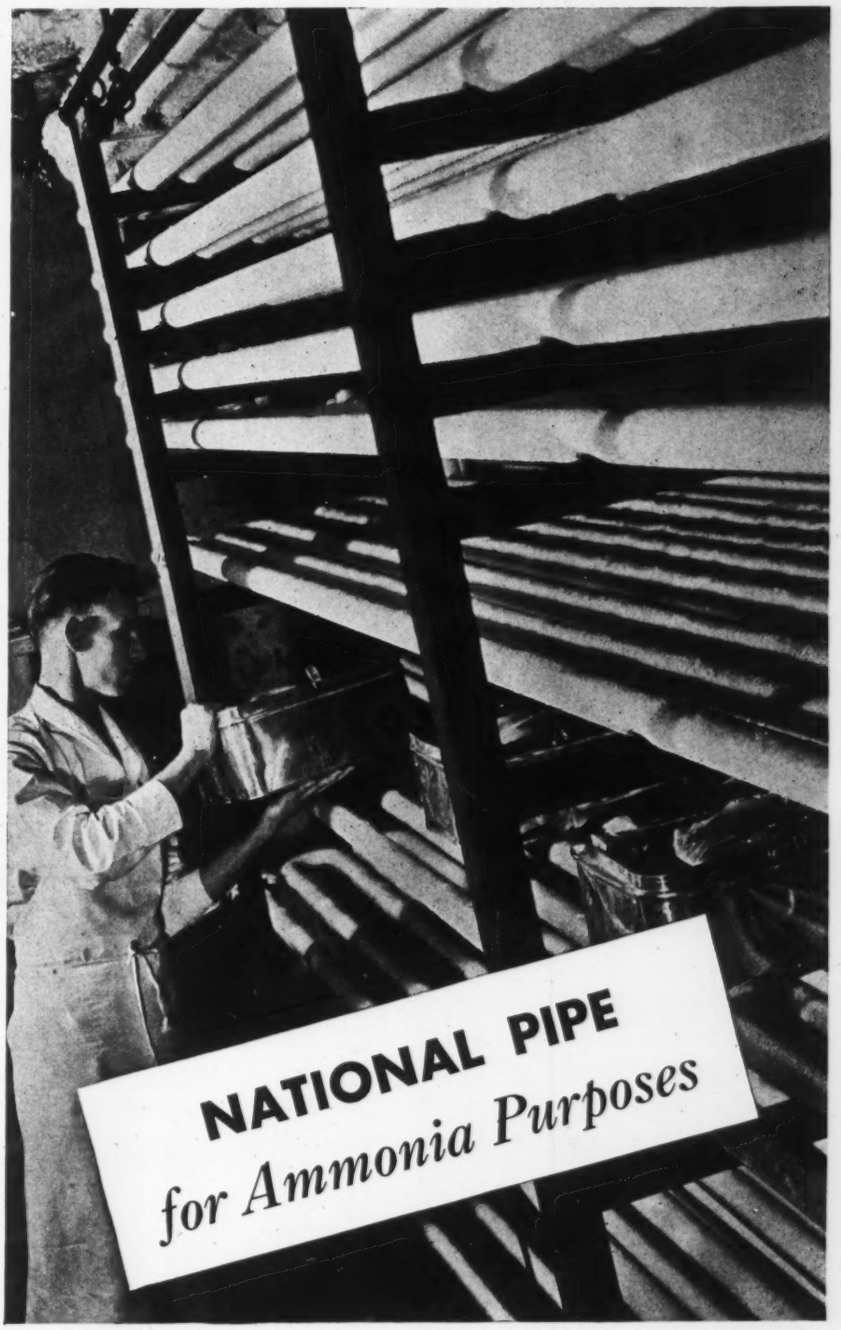
Fig. 2—Outside design wet bulb temperatures for cooling estimates

Table 4—Inside Design Conditions for Comfort Cooling

Outside Design Dry Bulb	Class AA Special Application Occupancy Over 40 Min.			Class A Average Application Occupancy Over 40 Min.			Class B Occupancy Under 40 Min.		
	Dry Bulb	Wet Bulb	R. H. %	Dry Bulb	Wet Bulb	R. H. %	Dry Bulb	Wet Bulb	R. H. %
85	73	64	64	74	64	60	75	64	57
	74	62	53	75	62	50	76	63	48
	75	61	45	76	61	43	77	61	41
	76	59	38	77	59	35	78	59	32
90	75	66	65	77	67	62	78	68	60
	76	65	57	78	66	53	79	66	53
	77	63	48	79	65	48	80	65	45
	78	62	41	80	63	40	81	64	40
95	77	67	60	79	68	58	81	69	55
	78	66	54	80	67	51	82	68	49
	79	65	47	81	66	45	83	67	44
	80	64	42	82	65	40	84	66	38
100	78	69	65	80	69	58	82	70	57
	79	68	57	81	68	52	83	69	51
	80	66	50	82	66	44	84	68	45
	81	65	43	83	65	38	85	67	38
105	79	69	63	81	70	58	83	71	57
	80	68	57	82	68	51	84	70	50
	81	67	49	83	67	45	85	69	45
	82	66	43	84	66	39	86	68	40

Note: It is recognized that in certain unusual localities, the outside design dewpoint is substantially less than the dewpoint of the tabulated inside design conditions. In such cases, the code committee will entertain the recommendations by local authorities of the inside design conditions for use in those localities.

COLD FACTS . . . *about Dependability and Ease of Fabrication*



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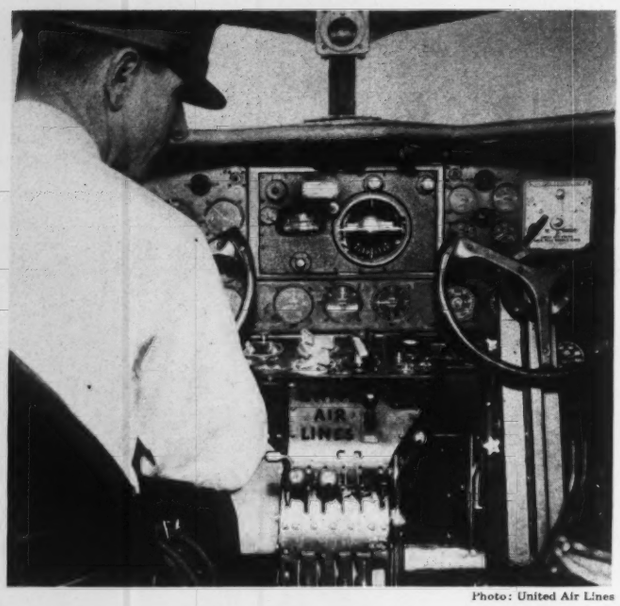

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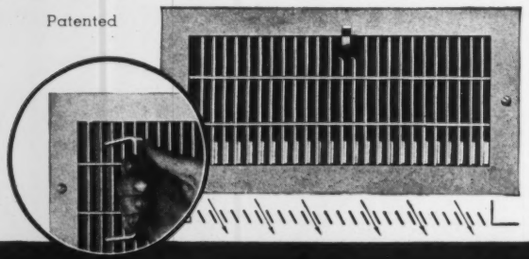

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able. If the application of the above formula to au thoritative weather records for an individual localit determines outside design conditions which differ fro those given by the map, they will be approved instead. These design outside conditions are considered to occu at 3 p. m.

F. Design Inside Conditions

1. *Heating*
   a. The design inside dry bulb temperature shall be 70 F.
   b. The design inside relative humidity shall be 35 per cen with an outside temperature of 30 F.
2. *Cooling*
   a. The design inside temperature and humidity shall be not more than that shown in Table 4.

G. Heat from Sunshine (Basis of determination to be added later).

H. Design Outside Air Quantity—The total quantity of outside air used as a basis for design load calculations shall be the infiltration, or the air required for ventilation, whichever is

Table 5—Door Infiltration in Summer Commercial Establishments

Application	Infiltration Cfm per Person in Room	
	72 in. Revolving Door	36 in. Swinging Door
Revolving and Swinging Doors Opening to Outside		
Bank	6.5	8.0
Barber Shop	4.0	5.0
Broker's Office	5.5	7.0
Candy and Soda	5.5	7.0
Cigar Store	20.0	30.0
Department Store (small)	6.5	8.0
Dress Shop	2.0	2.5
Drug Store	5.5	7.0
Furrier	2.0	2.5
Hospital Room	....	3.5
Lunch Room	4.0	5.0
Men's Shop	2.7	3.7
Office (Private)	....	2.5
Office (Professional)	....	3.5
Restaurant	2.0	2.5
Shoe Store	2.7	3.5
When Doors are Left Open Continuously		
72 in. Revolving Door (panels open)	1200 cfm	
36 in. Swinging Door (standing open)	800 cfm	

Note: These values for swinging doors and for doors left open hold only where such doors are in one wall only, or where the doors in other walls are of the revolving type. If swinging doors are used for access, or doors are left open, in more than one wall, the infiltration cannot be estimated. The values for revolving doors hold regardless of number or location.

To determine the total cfm infiltration due to opening of doors, multiply the design number of occupants by the factor from the above table for the kind of establishment in question. When there is more than one door, treat them as though there were only one, except in case of open doors.

Table 6—Infiltration Through Cracks Around Windows

Type of Window	Remarks	Cfm/Ft of Crack Wind Vel., 10 Miles per Hour		Cfm/Sq Ft of Entire Sash Area, Wind Vel. 10 Miles per Hour	
		Single Window	Double Window or Storm Sash	Single Windows	
				Small	Large
Double Hung Wood Sash Windows (unlocked)	Avg. window not w'strpd.	0.36	0.28	0.46	0.29
	Avg. window weatherstrip'd	0.27	0.20	0.34	0.22
	Poorly fitted window not weatherstripped	1.16	0.85	1.50	0.93
	Poorly fitted window w'strpd	.33	0.27	.42	0.27
Double Hung Metal Windows	Non-weatherstripped (unlocked)	0.78	0.60	1.00	0.62
	Weatherstripped (unlocked)	0.33	0.27	0.42	0.27
Single Sash Metal Windows	Hollow metal (vertically pivoted)	1.75	1.48	.....	.....
	Residential casement	0.55	0.40	.....	.....

Infiltration through door cracks may be assumed to be twice that of window cracks. Small windows are approximately 30 in. x 72 in., large windows approximately 54 in. x 96 in.

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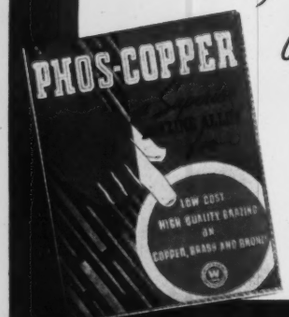

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greater. These shall be determined as outlined in Paragraphs I and J below.

I. Infiltration

1. *Heating*—Infiltration rates for calculating heating loads shall be determined in accordance with the latest edition of the *A.S.H.V.E. Guide.*

2. *Cooling*—Infiltration rates for calculating summer cooling loads shall not be less than that indicated from Tables 5 and 6.

J. Ventilation Requirements

1. In no case shall ventilation be less than that required by any local ordinance.

2. For summer cooling, the ventilation per stated occupant shall be not less than that shown in Table 7.

K. Design Occupancy

1. The design load calculations shall be based on the stated occupancy of the building during the time of maximum design conditions.

2. The heat given off by each occupant shall be calculated as not less than that in Table 8.

L. Heat from Appliances—(A table is to be included later).

M. Transmission Coefficients—Heat conduction shall be based on transmission coefficient for physical barriers such as walls, doors, windows, ceilings, floors, etc., as given by the 1932 edition of the *A.S.H.V.E. Guide.*

N. Total Air Circulation—The total air circulation is not specified. It shall be determined by the heating and cooling load and the type and arrangement of supply openings. It shall be adequate to meet the requirements under O.

O. Air Distribution

1. The quantity and temperature of the treated air and the method of introducing it to the conditioned space shall be de-

Table 7—Ventilation Standards

Application	CFM per Person	
	Preferred*	Minimum*
Apartment†	15	10
Banking space	10	7½
Barber Shop	10	7½
Beauty Parlor	10	7½
Broker's Board Room	30	20
Cocktail Bar	20	15
Department Store	7½	5
Directors' Room	30	30
Funeral Parlor	10	5
Hospital Room†	15	10
Hotel Room	15	10
Office, General	15	10
Office, Private†	30	15
Restaurant	15	12
Shop, Retail	10	7½
Theatre	7½	5
Note: For General Application		
Each person, not smoking	7½	5
Each person, smoking	30	25

†Figure at least one person for each 50 ft of floor area, but do not pyramid such loads for multiple rooms beyond the maximum simultaneous peak.

*Whether the preferred or minimum values are used, the outside air ducts shall be sized to admit at least 50 per cent more than the preferred values.

Table 8—Heat of Persons, Btu per Hr

Design Room Temp.	Activity Class A — Per Person Sitting or Moving Slowly			Activity Class B — Per Person Working or Dancing		
	Sensible Heat Btu per Hr	Latent Heat Btu per Hr	Total Heat Btu per Hr	Sensible Heat Btu per Hr	Latent Heat Btu per Hr	Total Heat Btu per Hr
84	180	220	400	150	510	660
82	200	200	400	180	480	660
80	220	180	400	210	450	660
78	240	160	400	240	420	660
76	256	144	400	270	390	660

Table 9

Air Current Velocity Linear Feet per Minute	Temperature Difference Average F Minus Current F
40 and over	2
Less than 40	3

signed to limit, to 3 F or less, the variation in dry bulb temperature at the same level throughout that portion of a single room that is normally frequented by persons.

2. It is desirable to avoid air velocities exceeding 50 lineal feet per minute in the zone between the floor and the 5 ft level, in spaces normally frequented by persons who are not normally moving about. Exception must be made of the vicinity of a supply or return grille when construction necessity requires it to be located below the 5 ft level and in a space normally frequented by occupants.

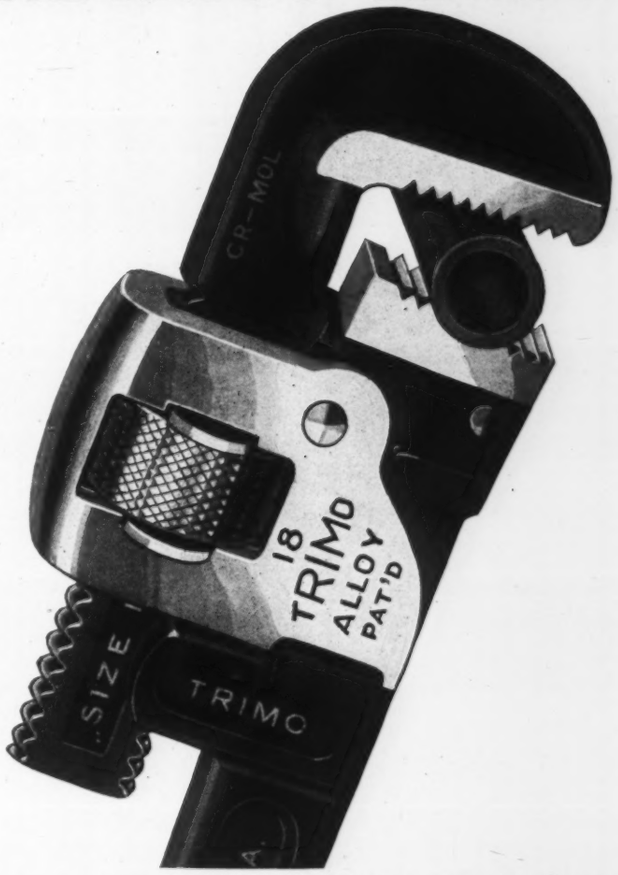
3. For cooling, it is desirable that the difference between the temperature of air currents in the space frequented by occupants and the average temperature in such space be not greater than as shown in Table 9 for the various air current velocities.

P. Duct Capacity—It is desirable that return ducts be sized large enough to handle 100 per cent of the circulated air to permit economical heating and cooling prior to occupancy. It is desirable that the outside air duct be large enough to handle 100 per cent of the total circulated air to permit economical operation during mild weather, but in every case it shall be large enough to admit 50 per cent more air than that determined from the preferred values in Table 7.

Q. Capacity Specifications—The air conditioning equipment shall have a capacity equal to the design load defined in Paragraph D under the following conditions:

1. *Cooling*
   - a. Design outside conditions defined in Sec. E-2.
   - b. Design inside conditions defined in Sec. F-2.
   - c. Design air quantity defined in Sec. G.
   - d. Design occupancy defined in Sec. J.
   - e. Condensing medium for refrigeration equipment.
     - (1) Water cooled refrigeration equipment.
       - (a) City water temperature not less than the average maximum value for the months of July and August. (*A.S.H.V.E. Guide*)
       - (b) Well water temperature not less than the average maximum value for the months of July and August.
       - (c) Cooling Tower—The tower shall deliver water not more than 10 F above the design wet bulb temperature.
     - (2) Air cooled refrigeration equipment. The condenser air temperature shall be not less than the design dry bulb. Allowance should be made for temperature rise above ambient for an air cooled condensing unit installed in a room, its amount depending on the unit size, room volume and rate of ventilation.
     - (3) Refrigeration equipment with evaporative condenser cooling. The condenser shall be selected for a wet bulb temperature not less than the design wet bulb. Allowance should be made for temperature rise above ambient for an evaporative condenser installed in a room, its amount depending on the unit size, room volume and rate of ventilation.
2. *Heating*
   - a. Design outside conditions as defined in Sec. E1.
   - b. Design inside conditions as defined in Sec. F1.
   - c. Design outside air quantity as defined in Sec. H1.
   - d. Heating medium at a temperature that can be maintained by the heat generating device when delivering heat to the heating device at a rate not less than the design heating load.

ALWAYS READY



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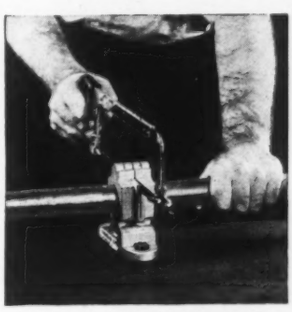

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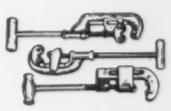

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R. Noise

1. The sound level meter specified by the *American Standards Association's* committee on sound, shall be used for noise readings. Noise values to be given in terms of db, based on the 40 db weighted network.
2. Where noise specifications are made they shall specify the room noise level due to the operation of the apparatus corrected to the noise level of the room when apparatus is not in operation.
3. Noise readings in the conditioned space shall be taken—
   a. In the case of apparatus located within the room, at a height of 5 ft and at a distance of 5 ft from the apparatus.
   b. In the case of air inlet or outlet openings, at a height of 5 ft and in front of the opening at a horizontal distance of 5 ft from wall in which opening is located.

Booklets, Reports and Papers

Steam Jet Refrigeration for Air Conditioning

Too much emphasis cannot be placed on the steam jet cooling unit as a summer load builder for district steam companies, according to G. K. Saurwein, chairman of the research committee of the *National District Heating Association*. The committee's report, presented at the *N.D.H.A.* convention last month, was consequently devoted almost entirely to the subject.

M. A. Nelson of the Westinghouse Electric & Mfg. Co. outlined the economic application of steam jet units, tracing early developments and listing a few of the improvements made in this type of equipment. Three types of condensers may be employed; they are the surface, low level, and barometric. Among the advantages of steam jet machines are the use of water as the refrigerant, very few moving parts, small weight per unit of floor space, and high partial load efficiencies obtained at as many points as there are booster ejectors. In those systems where an open spray washer is used in series with the vacuum cooler, smoke and objectionable odors absorbed by the water are given up in the high vacuum and are removed by the secondary ejectors, the water returning to the sprays thus being reactivated.

Cooling water requirements and operation of steam jet units were compared with other types of refrigeration. A fundamental difference of the steam jet unit is that as the motive steam pressure decreases, a point is reached at which the booster "breaks" back; that is, the compressor suddenly drops all its load. Other types of refrigeration would not drop the load but would continue to operate at reduced capacity.

Operating difficulties which sometimes arise were listed, and the economic application of this type of equipment was discussed. Practical limitation for the operating pressure is about 2 lb gauge, with steam consumption materially decreasing with increasing pressure up to 100 lb, above which the gain in steam economy is slight.

The second paper of the report, by A. A. Markson and R. Menke of the New York Steam Corp., discussed the testing of steam jet coolers in winter under artificial loads. As the steam jet booster is a device for removing vapor from the flash tank and compressing it to condenser pressure, the measured direct introduction of live steam into the flash tank would logically provide the vapor for the jets to remove and thus load the machine. Therefore, a flow meter was connected to the flash tank so that metered steam was expanded directly into the space above the water level in the flash tank.

Capacity tests were made as follows: The machine was placed in operation with the spray water being circulated through a bypass. The live steam valve was opened until the flow meter indicated the desired flow, the approximate tonnage capacity based on the number of boosters being estimated and calculated in terms of steam flow into the flash tank for loading the machine. Readings were taken of booster steam pressure, con-

denser absolute pressure, chilled water temperature in and out of flash tank, and condenser water temperatures, the readings being recorded as test points after the flash tank temperature reached a steady value. A set of such readings gives one point on the output versus temperature characteristic curve of the machine. The steam loading was then varied in both directions, each change in tonnage giving a point on the characteristic curve. Booster steam consumption was calculated by the use of Grashof's formula for steam flow through nozzles, throat diameters being gaged to 0.001 in.

Curves for tonnage output and unit steam consumption of single boosters, overall load characteristics, and pressure—stability characteristics were presented. It was the author's belief that the methods described were simple and were superior to tests conducted by taking very small temperature differences and measuring chilled water flow under the varying loads of actual summer conditions.

Control by Water Flow Measurement

The third paper of the report (by J. M. Sperzel, Engineer, Rockefeller Center, New York City) described the development and application of the method of control for the steam jet air conditioning systems at the British Empire building and La Maison Francaise, Rockefeller Center, the problem being to operate the plant at reasonable economy consistent with varying load conditions with as little attendance as possible, which was partially solved by substituting the rate of water flow as a control source for the refrigerating effect, in place of the temperature of the chilled water, as is usual. The system comprises a steam jet machine connected to a number of dehumidifiers arranged in parallel.

The author showed that the steam jet machine improves in thermal efficiency with higher chilled water delivery temperatures, and that substantial savings can be effected, load conditions permitting, if the delivery temperature is allowed to rise with decreasing loads. Hence, by holding the delivery temperature constant independent of changes in load, economy is sacrificed at light loads.

The control devised consisted of replacing the delivery water temperature measurement by a flow measurement as a source of control for the intermittent operation of the automatic ejector, the control consisting of an orifice plate inserted in the delivery piping of the machine, and a specially constructed mercury manometer possessing three electrodes, two of which were inserted in the low pressure leg and the third located so as to be totally submerged in the mercury. The position of one of the electrodes in the low pressure leg is set, so that a particular deflection of the mercury column will cause an electrical circuit to be completed to start the automatic ejector, when the pressure difference across the orifice plate reaches the corresponding predetermined value of flow. Once started, the electrical circuit is arranged to maintain the ejector in operation until such time as the circuit is opened by the second electrode breaking contact with the mercury column. This occurs when the deflection corresponds to a lower predetermined value of flow. Consequently, the operating time of the ejector is governed by the time it takes the flow of chilled water to fall from the upper to the lower predetermined limits of flow. Likewise, the idle time of the ejector is a function of the time of flow rise between the predetermined limits.

The author concluded that in this case flow control reduced the instantaneous measurement of economy from the guaranteed minimum value of 22 lb to 18 lb of steam per ton-hour, and over the summer period the total steam consumption was reduced approximately 15 per cent.

Industrial Dust Control

"Industrial Dust Control Through Exhaust Systems," is the title of a recently published booklet by W. O. Vedder. Not a text book on dust control, its object is rather to give the plant

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● Is it made of high quality materials—by a process that insures absolute uniformity; true size; perfect roundness; tight welds; straightness; sharp, accurate threads; unvarying ductility in every inch; freedom from hard or burnt spots in the metal; freedom from scale; cleanliness inside and outside; ease in bending, cutting and threading; and a galvanized zinc alloy surface that resists rust and abuse in handling? Is it available in long lengths without cross-weld? • If you want all of these qualities in the steel pipe you use, you want Fretz-Moon Pipe—pipe made by the exclusive "continuous process" that takes all guesswork out of pipe manufacture—pipe that cannot vary in its high quality. • The complete story of modern pipe making by the "continuous process" is contained in an illustrated booklet. Write for a copy.

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executive a general descriptive outline of the means of controlling industrial dusts at the source of production through the medium of exhaust systems.

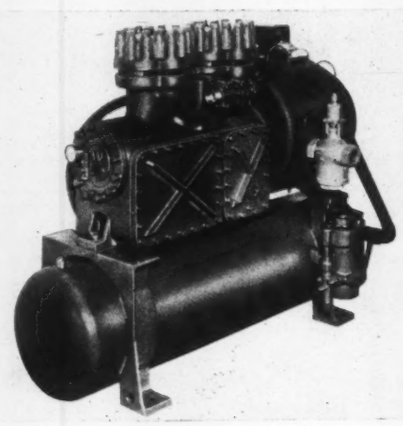
Among the subjects discussed are exhaust hoods and piping systems, dust collecting equipment, types of collectors, the exhauster and drive, and operation and maintenance of exhaust systems. Copies of this 50 page, pocket size booklet are available without charge, for a limited time, from the Pangborn Corp., Pangborn Blvd., Hagerstown, Md., if request is made upon company letterhead.

Equipment Developments

For your convenience in obtaining more information about any of this equipment, see coupon on page 130. Add the new products and companies listed here to your Directory Section which you received in your January 1936 Heating, Piping and Air Conditioning and thus keep your records of sources of supply up to date throughout the year . . . Single asterisk (*) indicates equipment not listed in Directory Section; double asterisk (**) equipment and manufacturer not listed.

Hermetically Sealed Air Conditioning Condensing Units

No. 1015—New line of hermetically sealed air conditioning condensing units (Type "CLS") with nominal ratings from 7 to 22 tons cooling capacity announced last month have motor completely enclosed with the compressor, eliminating the shaft seal and preventing leakage of refrigerant, or dust, dirt and moisture from reaching working parts. The rotor of the driving motor is overhung from the main rear compressor bearing on a short, stubby extension of the compressor crankshaft; this gives a compact direct drive and avoids ever having to oil the motor and eliminates use of belts and pulleys. The motor is water cooled, making the unit suitable for installation in unventilated locations. Units are therefore compact, light in weight and require less maintenance.



A simple oil pump positively distributes the oil to every part of the unit. Direct driven from the main crankshaft by a noiseless Micarta gear, the pump consists of two rotating gears which pump the oil from the reservoir, strain it through a fine mesh metal screen, and deliver it through rifle-drilled passages in crankshaft and connecting rods to all bearing surfaces. Oil pumped with the refrigerant is removed from the refrigerant by an oil separator with no moving parts, is strained, and returns to the oil reservoir for recirculation.

Crankshaft is harmonically balanced, helps make the unit almost noiseless and so free from vibration that a coin will stand on edge on top of the unit when running, according to the manufacturer.

Series parallel condenser manifold permits efficient use with

either city water or cooling tower, allows the selection of pressure drop through the condenser by placing two sets of coils in either series or parallel to best suit local conditions. A pilot operated water valve is used for control.

Mechanical adjustments may be made through side service plates. The entire operating mechanism of the unit can be exposed by removing the side plates from the crankcase casting. No refrigerant or water lines need to be disconnected to service the interior of the unit, and the unit itself is not disturbed in any way. Servicing is thus a simple operation, entailing only the removal of the bolts holding the plates in position, and their being tightly screwed up again when adjustments have been made. Even the oil pump is accessible. Major parts are interchangeable, resulting in quick service and simplified stocking of repair parts.

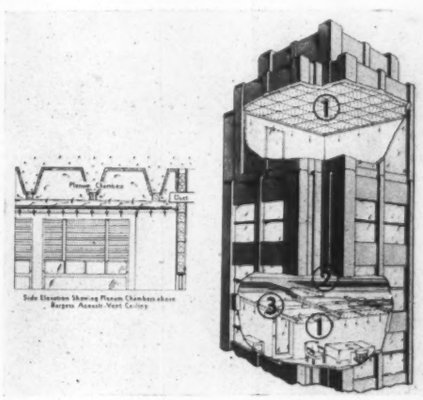
Two, four, and six cylinder units are available, with connected motor hp of 7½, 15, and 25 hp, nominal ratings 7 to 22 tons. The 14 ton unit is illustrated.—Westinghouse Electric & Mfg. Co., Mansfield, Ohio.

New Air Distribution and Sound Absorption System

No. 1016—A new system of air distribution has been devised to solve, with one type of construction, the dual problem of air circulation without drafts and the quieting of occupational room noises.

Known as the "Acousti-Vent," this system comprises a perforated ceiling made of a suitable architectural surface installed slightly below the normal ceiling level of the room. Between the perforated sheet and the room ceiling is a sound absorbing material that is installed with sufficient clearance so that a space is provided between this sound absorbing material and the ceiling. This space provides a plenum chamber into which air is introduced through ducts from the ventilating fan.

A uniform, low static pressure is maintained in this plenum chamber and suitable means are provided to cause the air to pass to the underside of the sound absorbing material without passing through it, and thence through the myriads of small openings in the perforated sub-ceiling into the room.



Air flows imperceptibly and uniformly through the perforated ceiling into every part of the room, affording accurate temperature control, states the manufacturer. Tests indicate that there is no appreciable temperature variation throughout the room. Due to the large ceiling area and the low air velocity at every point, this system makes possible a rapid change of heated or cooled air without drafts.

Second major feature of the system is its efficiency as a sound absorbent. Room noises, upon reaching the perforated metal ceiling, seep through the small perforations and encounter the resilient sound absorbing material placed behind the perforations. These sound waves are then absorbed, causing a marked reduction in the noise level of the room. In addition to absorbing room noises, noises of the ventilating system are absorbed.

The system is applicable to air conditioning installations in office buildings, hospitals, restaurants, auditoriums, and other places where the dual problem of ventilation and acoustic treat-

Akron Savings & Loan Building, Akron, O.

MODERNIZE HEATING IN AKRON BANK BUILDING

Webster Hylo System Is Selected After Exhaustive Investigation

Akron, O.—The engineering experience of the Webster Organization was a determining factor in the selection of the Webster Hylo System of Steam Heating and Webster Heating System Equipment in the Akron Savings & Loan Company Building, according to Building Manager A. B. Crandall.

The heating system in this 12-story office building was brought up-to-date during the 1934-35 heating season after an exhaustive investigation of heating control methods.

"While our primary object in installing the Webster Hylo Control System was to obtain even temperature and tenant comfort, it is interesting to note that a 19 per cent coal saving was also effected."

The Webster Hylo Steam Variator—a simple, rugged central control for use with Webster Systems where distribution of steam to all radiators is balanced by accurately sized metering orifices. The Hylo Variator Valve Operator (illustrated) automatically throttles a rugged motor-operated valve in the steam main.

How this 19 per cent saving was achieved is shown in the performance record below. There was no change in the heating schedule of 91 hours a week.

Heating Performance Record

	Degree Days	*Lbs. Coal Fired*	*Steam Per D.D. Per M. Sq. Ft. Radiation*
Before Control 1933-34.....	6,566	875,000	79.5
After Control 1935-36.....	6,728	730,000	64.5

Data for season 1934-35 is not conclusive, as change was made during the heating season.

If you are interested in heating *new* buildings, or in improved heating service and lower heating cost in your present building, address

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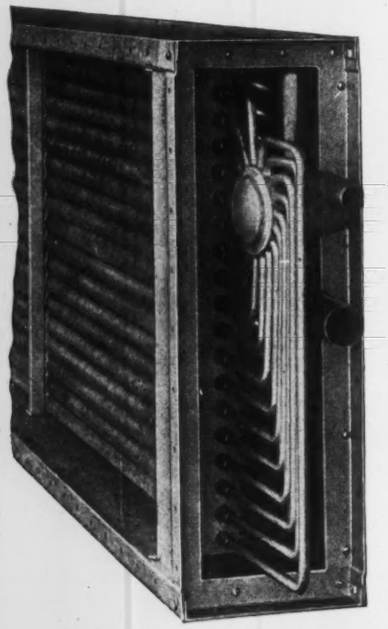
Announcing

AEROFIN

Direct Expansion Cooling Coils with Exclusive Centrifugal DISTRIBUTING HEADER



Aerofin Direct Expansion with new Centrifugal Distributing Header, designed to meet changing conditions in air conditioning practice.

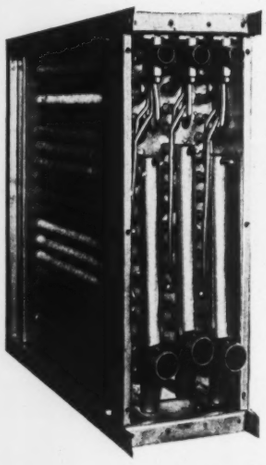
Below: Aerofin Direct Expansion with standard distributing and suction headers allowing control for each row.

THE new Aerofin Centrifugal Distributing Header for direct expansion cooling coils is the result of extended research and extensive field tests.

It provides a proven method for automatically distributing the refrigerant evenly through all circuits of the evaporator where only one connection for each coil is required. It saves costs on installation, is simple and fool-proof.

This distributing header, like all standard Aerofin equipment, minimizes pressure loss. Made in 2 to 6 rows of tubes in standard 10" depth casings.

New Supplement to Bulletin D. E. 34, gives complete tables and curves for selecting proper size and type. Permits accurate determination of final air condition leaving coil. Send for your copy today.



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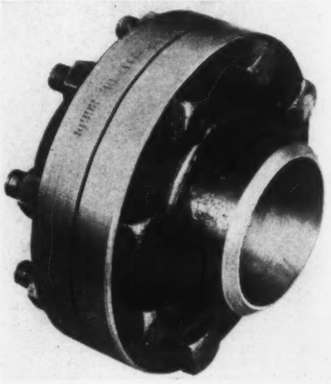
ment must be solved. The office building installation illustrated shows (1) the perforated ceiling; (2) air supply duct; and (3) sound absorbing pads.— Burgess Battery Co., 111 W. Monroe St., Chicago, Ill.

Steel Welding Flanges for Refrigeration Piping

No. 1017—Because the welding of refrigeration piping has become popular, especially for air conditioning installations, a new line of steel welding flanges has been developed. Made entirely of forged steel, the new flanges, capable of withstanding a much greater pressure than would ever be found in a refrigerating plant, are of the tongue and groove type, and have practically the same cross section as the maker's standard screwed flanges.

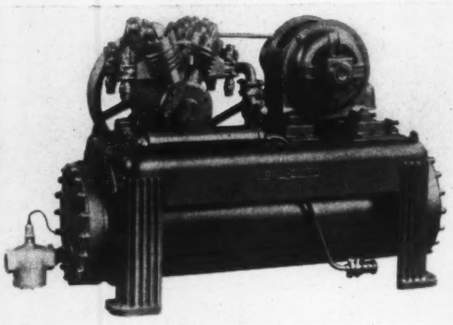


The flanges have an extra long welding neck to insure (1) plenty of room for the welding operator, (2) free flowing metal, insuring a perfect weld, (3) diffusion of heat, preventing warping of the flange, and (4) liberal clearance for tightening the tie-bolts. The flanges are made in oval, square, and round types, in a range of pipe sizes from ¾ to 10 in.

The company has also designed pipe welding plugs for use as a fixture to hold flanges in line with the pipe during the welding operation; these are of solid copper alloy which readily diffuses the heat of the weld, and does not adhere to the molten steel.— York Ice Machinery Corp., Roosevelt Ave., York, Pa.

V-Type 40-Ton Air Conditioning Compressors

No. 1018—New air conditioning refrigeration machine embodying many of the principles of design used in automobile and airplane engines features a V-type compressor of eight cylinders. Due to their efficient design, utilizing a large capacity shell and tube type of condenser, high economy of water and electric power has been obtained, states the maker. The V-type compressor,



electric motor, and condenser are all mounted in one compact unit, which has a refrigeration capacity of 40 tons.

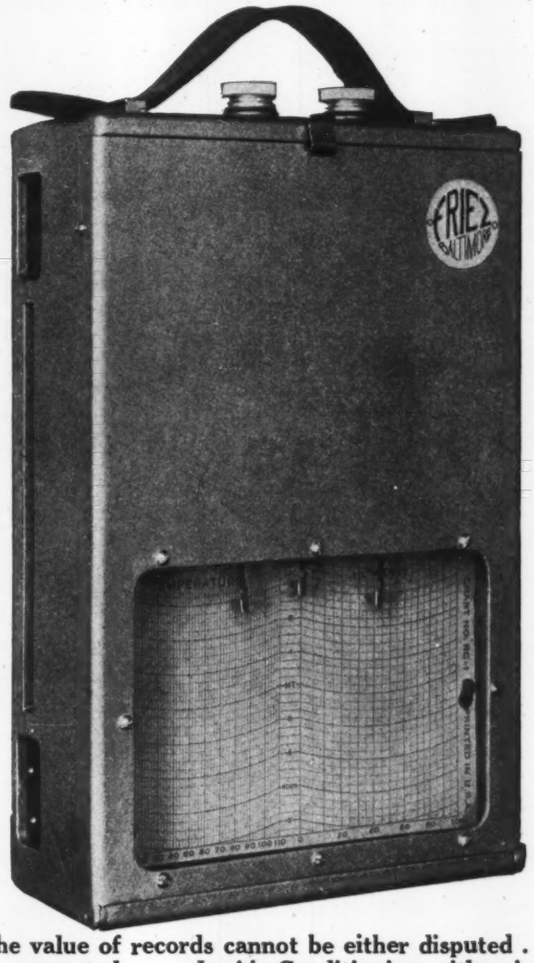
By using two or more of these condensing units on large installations in theaters, stores, or hotels it is possible to obtain flexibility and accuracy of control.—General Electric Co., Bloomfield, N. J.

New Rods for Light Gauge Welding

No. 1019—"Thinweld," a specially coated electrode for arc welding light gauge sheet metal, is now offered in $\frac{1}{16}$, $\frac{3}{32}$ and $\frac{1}{8}$ in. sizes. Among its light welding applications are the fabrication of heating and air conditioning cabinets, ducts, etc. It is reported that metal as light as 24 and 26 gauge has been successfully welded with this rod.—Hobart Bros. Co., Troy, Ohio.

FRIEZ BALTIMORE

Let's Look at the Record



The value of records cannot be either disputed . . . . or computed . . . . In Air-Conditioning, either in industry or large scale comfort installations, a complete score of temperature . . . . humidity . . . . running time . . . . has become a necessity . . . . for engineering . . . . for sales . . . . for servicing. Compare all the Recording Instruments you have ever heard of . . . . the Friez Portable Recorder is the only one designed solely for all these specific purposes. It can be carried as easily as a camera . . . . its records are as accurate . . . . as legible as a photograph . . . . charts are of the size of filing cards and of the same shape. Temperature . . . . *Humidity . . . . Running Time . . . . a complete record on a portable, rugged recorder.

**Humidity is recorded directly without need of computations or tables.*

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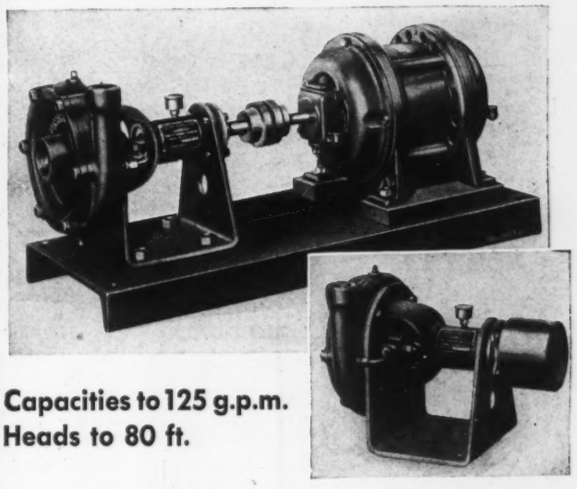
CENTRIFUGAL PUMPS

MODERN BALL BEARING UNITS FOR A WIDE RANGE OF USES



Capacities to 125 g.p.m.
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Check these features:

- Pressed steel frame mounting
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HIGH quality combined with low cost . . . the product of long experience, engineering skill, expert workmanship and advanced manufacturing methods.

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The one excels because of its close approach to absolute classical perfection—the other because of its performance record for two generations.

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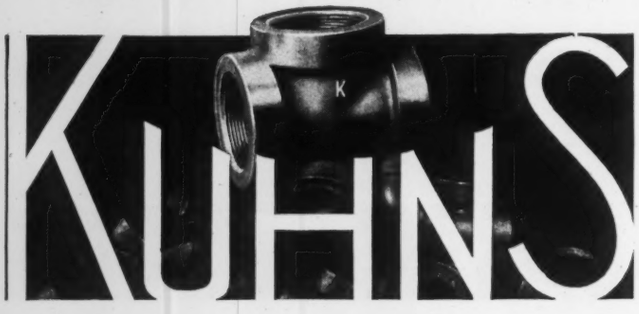
Catalog available for Plumbers and Industrial Users.

KUHNS BROTHERS COMPANY
Dayton, Ohio **1887**



Reversible Ratchet Wrenches Announced

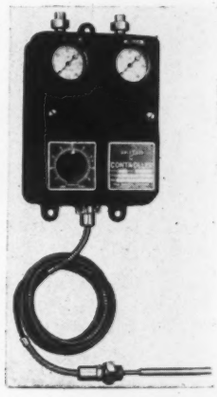
No. 1020—New heavy duty reversible ratchet wrenches ("Superector") embody unique structural and design features. Outstanding among them are quadruple pawls to provide double bearing and strength with increased durability. Handles are

drop-forged, to utilize the extra strength afforded by the pawls.

Built for rapid operation in the most severe service such as bridge, structural and other work requiring strong, fast-action tools, the wrenchers are made in 5 sizes, 24 to 48 in. Both hex and square sockets, with hole extending clear through, openings $1\frac{1}{16}$ to $4\frac{5}{8}$ in., turn nuts on any length of bolt.—J. H. Williams & Co., 75 Spring St., New York, N. Y.

Pneumatic Type Controller

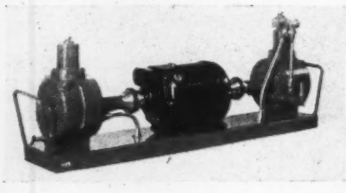
No. 1021—New pneumatic-type controller for temperature, pressure and liquid level, known as the "Model 90 Controller," has been developed to take care of the requirements for automatic temperature and pressure control that do not warrant a chart record of the controlled temperature or pressure, and also, for applications where recording instruments already are in use. Feature of the controller is its precision measuring element, of the same type as employed in the manufacturer's recording instruments.



The controller is equipped with a wide-range sensitivity adjustment. It can be changed from direct to reverse acting, or *vice versa,* by the user without additional parts. The case is moisture, fume, and dust proof and is offered in two forms, for wall and for flush panel mounting.—Bristol Co., Waterbury, Conn.

Vacuum Pump for Testing Refrigerating Lines

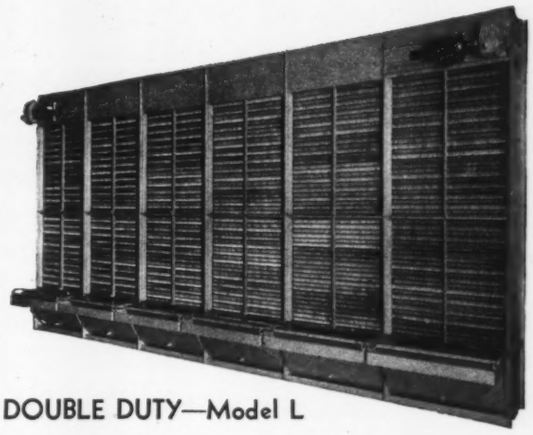
No. 1022—New vacuum pump has been developed to give economical and dependable service in testing air conditioning systems. Pump is used for evacuating refrigeration systems prior to introducing the refrigerant, for testing the tightness of refrigerating lines before putting an air conditioning system in service, and for testing for leaks on installations where trouble has developed.



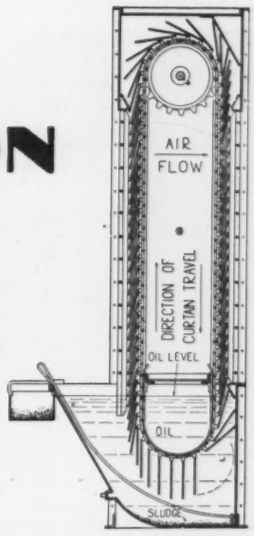
Unit is portable, and is mounted on a base with two welded handles. There are two vacuum units, each direct connected to a motor located between them. These two units may be operated in series (compound) to produce a vacuum within ½ mm of absolute, or in parallel (duplex) to produce a vacuum within 1½ mm. It is furnished in two sizes with a capacity of 1 cfm (compound) and 2 cfm (duplex) and 2 cfm (compound) and 4 cfm (duplex).—Beach-Russ Co., 50 Church St., New York, N. Y.

True IMPINGEMENT—Air FILTRATION

The "DOUBLE DUTY" Model L air filter, illustrated on the right is particularly suited to heavy duty, large capacity, general ventilation installations in large buildings and industrial applications demanding strictly automatic operation. It's sturdy construction guarantees unlimited years of continuous, trouble-free performance. The Model L is built in single units of 5000 cfm. and may be operated in combination groups to meet any requirements.



DOUBLE DUTY—Model L



Cross-section thru filter, showing curtain operation, each curtain offering three acute air deflections.

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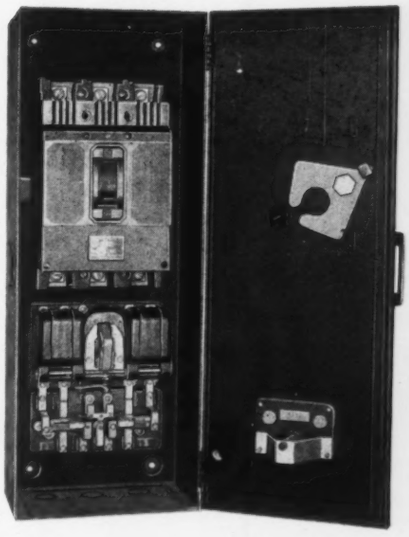
Clark-Sundh

A Clark Bulletin 6020 Across-the-Line Type Automatic A.C. Starter with externally operated Air Circuit Breaker, eliminates the usual separate fused safety switch. The air circuit breaker serves as a disconnect switch and also provides short circuit protection.



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TAKE THEM OUT *with*
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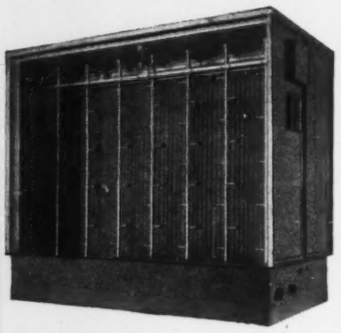
Makes air as cool and clean as mountain mist. Cleanses, cools, humidifies, de-humidifies. Sizes and types to fit all requirements. Quiet in operation. An unusual combination of quality, efficiency and economy.



The National Line includes ventilating and air conditioning equipment for every need at attractive prices. We have a 100% agency policy—write for available territory.

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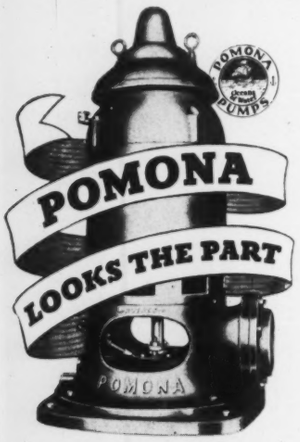


Pomona Pumps have a big part to play in the world's production of water. In every civilized country on the globe, men are looking to Pomona Pumps for their water supply under every type of condition.

Here is a pump that has never failed them. Pomona's look of durability—of quality—of efficiency—has not been paint deep; it has taken abuse and over-work and kept right on building a reputation for long life—for low operating expense—for trouble-free operation. Pomonas look the part and justify every confidence placed in them by their makers and users.

Send today for a brand new bulletin featuring things you should know about hydraulic laboratory tests.

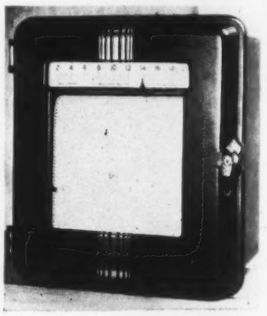
POMONA PUMP CO.
53 W. Jackson Blvd., Chicago • 206 E. Commercial St., Pomona, Cal.
POMONA TURBINE PUMPS

Coated Rods for D-C Welding

No. 1023—Line of coated rods for d-c welding includes five different types with both high and low rates of fluidity for various types of work in welding in flat, vertical or overhead positions and with ferrous and non-ferrous metals. Service tests show tensile strength of welds from 55,000 to 75,000 lb per sq in. with various types of rods ranging from 3/32 to 3/8 in. "Smooth-arc" electrodes are designed primarily to speed up welding operations with a smoother, more easily handled arc and to reduce spatter losses.—Harnischfeger Corp., 4400 W. National Ave., Milwaukee, Wis.

New Recorders and Controllers

No. 1024—"Silver Anniversary Micromax" recorder provides in one instrument all combinations of indicating, recording, signalling and controlling. Measurements are presented conveniently; a bold scale and pointer shows the condition at the moment, and a clear record on ten visible inches of strip-chart shows it for the past several hours. On controllers, a second pointer shows control setting. Multi-point records may be in blue or in multi-color.



Control and signal contacts operate undisturbed by air currents when the door is opened, when the chart is changed, or the pen refilled. Pen holds seven week ink supply. A double frame makes all parts of the mechanism accessible, and enables the user to replace the chart without effect on other parts of the recorder.

Available as an indicating recorder or as an indicating and recording controller for temperature, CO_2, smoke density, chemical strength, pH, frequency, remote load, etc.—Leeds & Northrup Co., 4952 Stenton Ave., Philadelphia, Pa.

Vertical Type Year 'Round Air Conditioners

No. 1025—Vertical type year 'round conditioners, designed to meet a wider range of air conditioning requirements, supplement regular line of horizontal units, are designed primarily for human comfort air conditioning applications such as department stores, restaurants, hotels, and various retail specialty shops, are used for conditioning single rooms, or groups of rooms from one unit. In the group system, several rooms with similar load characteristics are conditioned by a single unit, and in large installations several air conditioners may be operated in multiple from a central refrigerating machine.

Smallest of the new units has approximate 3 ton capacity, can be mounted either in horizontal or vertical position. It is provided with its own plenum chamber, lined with an acoustical material.

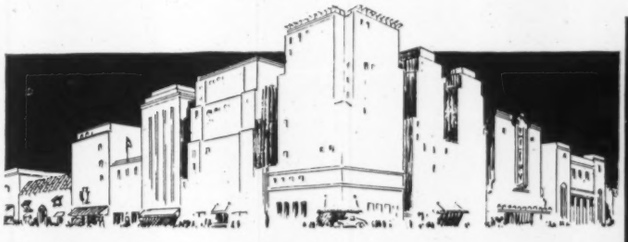
Night Club, Tap Room and Dining Hall of the Claridge Hotel, at Memphis.

All Air Conditioned

with FRICK

Refrigeration

Get your copy of the new Frick Bulletin on Air Conditioning. Write Frick Co., Waynesboro, Penna.



Wherever You Need Air Conditioning You Need Fedders Coils and Valves

Foremost Air Conditioning Engineers are specifying and installing Fedders All-Copper Air Conditioning Surface and High Capacity Thermostatic Expansion Valves. Factory Engineered Manifolding provides efficient distribution throughout all sections of the coil and eliminates necessity for field assembly. Thoroughly cleaned and dehydrated . . . tested with 300 pounds pressure . . . approved by the Underwriters Laboratories . . . cataloged in standard models for refrigerant, cold water or steam. Write for Bulletins giving complete data.

Heat Transfer Specialists for 40 Years

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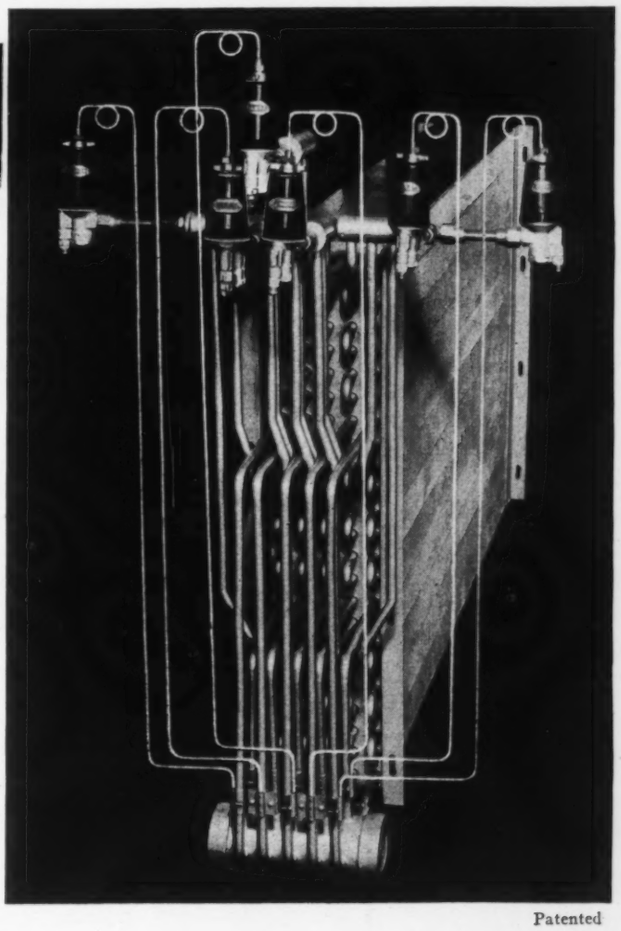

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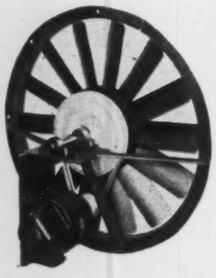
GENERALS
for
MAXIMUM PLANT EFFICIENCY

What is your problem? Controlling the flow of heavy oils? Live steam at high pressures? Refrigerants? Do you require close control of room temperature? Flow control in hot water heating systems?

Then look to GENERAL CONTROLS for the best answer. They will give you unequalled performance in precision control. Write for Catalog J-182.

GENERAL CONTROLS
1505 Broadway, Cleveland, Ohio — 1370 Harrison Street, San Francisco, Calif.
ORIGINATORS OF THE QUIET A. C. SOLENOID

THE DE BOTHEZAT
Vari-Speed
BELT DRIVEN FAN



Efficient, adjustable, variable speed fan for dwellings, theatres, clubs, etc.

- Provides 25% variation in speed, without electrical control, by manual adjustment of motor sheave. Adjustment easily made after installation to suit varying conditions.
- Fan and motor are mounted as a unit by mounting ring. No other support required.
- Motor will not overload. Has DeBothezat non-overloading power characteristics.
- Quiet in operation, rugged in construction, economical in power consumption.

Write us for descriptive bulletin and detailed information.

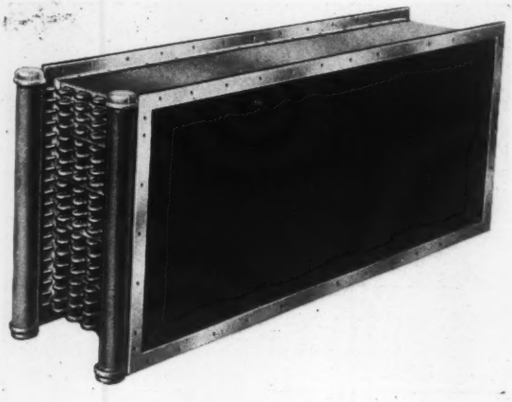
DE BOTHEZAT DIVISION
American Machine and Metals Manufacturing Corporation
100 Sixth Avenue New York, N. Y.

Larger units have approximate capacities of 6, 15 and 30 tons. They consist of casing, drip pan, fan assembly and motor, heating and cooling coils, thermostatic expansion valve, liquid distributing header, humidifier, and sectional air filter, all mounted within decorative casing of golden brown baked enamel.

A new feature is the use of variable pitch driving pulleys to permit close adjustment of fan speeds after installation. Cooling coils are corrugated fin type, for direct expansion refrigerant, cold water, or brine as the cooling medium. Heating surface is of the same type for either steam or hot water. All units arranged for bypass method of temperature control, when required.—York Ice Machinery Corp., Roosevelt Ave., York, Pa.

A. C. Coils Metal Dipped After Fabrication

No. 1026—New all-copper air conditioning coils with positive capacity ratings have permanent metal-to-metal contact between fin and tube obtained by hot tinning after fabrication and assembly. The permanent bond between fin and tube, obtained by dipping the completed coil, assures definite cooling rates, states the manufacturer. In sealing the contacting surfaces by metal dipping, oxidation of the contact points and entire copper surface is eliminated and the coils remain clean and afford high operating efficiency.



Sizes and capacities: Face areas from 1 to 10 sq ft up to twelve rows deep. Capacities at 85 F D. B., 50 per cent R. H. entering air, 500 fpm face velocity available from 6,000 Btu per hr and up. Coils also built to order for special installations with guaranteed ratings.—Standard Galvanizing Co., Air Conditioning Coil Div., 2611-2619 W. Van Buren St., Chicago, Ill.

Lamp Sterilizes Air, Tests Indicate

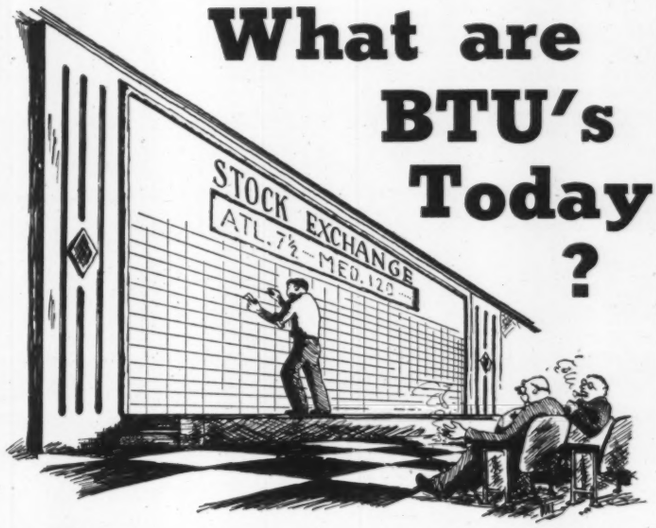
No. 1027—A new low wattage gaseous conductor device (the "Sterilamp") has been developed which produces radiations germicidal to mold spores in the air. Principal applications are in the food industry for killing mold spores and spoilage bacteria in refrigerator storage rooms, bakeries, etc. Tests have indicated that the lamp can also be used to sterilize conditioned air, says the manufacturer.

On a-c, the only necessary auxiliary equipment needed is a transformer. Three or four lamps may be operated from one transformer. The power taken by two lamps, plus that consumed by the transformer, is less than that required to operate an ordinary 25 watt incandescent lamp.—Westinghouse Lamp Co., Bloomfield, N. J.

Pressure Compensator for Flow Meters

No. 1028—In measuring flow of compressible fluids such as steam, air and gas, variations in the static pressure of the flow directly influence the reading. To prevent errors from this source, an automatic pressure compensator for flow meters has been developed. The mechanism is merely a spiral pressure tube connected to a lever.

What are BTU's Today ?



THE Big Board is not the only place where investments go up or down! Consolidated Power? United Fuel? What are BTU'S today? If BTU'S are down—look for leakage and loss in your underground steam lines. There is no "gambling" when your lines are enclosed in Ric-wiL Conduit—a safe investment.

Ric-wiL Conduit keeps underground steam lines tight, dry, and 90% or more efficient. Various designs in both standard weight Tile and Super Tile, in Cast Iron, and in Armco Iron Unit Steam Main. Famous Dry-paC Waterproof Asbestos Insulation, or other optional insulations. Ric-wiL offers a thoroughly engineered service, adequate to any problem in the field of sub-soil steam conveyance. Installation instructions, engineering service drawings, also field supervision if desired. Catalog promptly on request.

The Ric-wiL Co., Union Trust Bldg., Cleveland, Ohio
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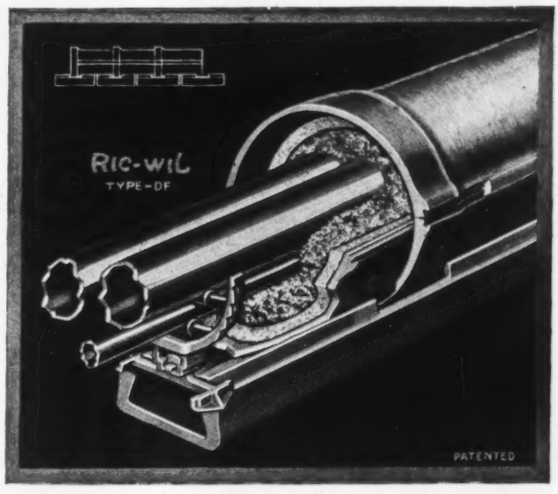
According to ASHVE Research Bureau, heat loss from conduit in wet soil is over *four times* that in dry soil. Ric-wiL External Base Drain keeps the soil *dry*.



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CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

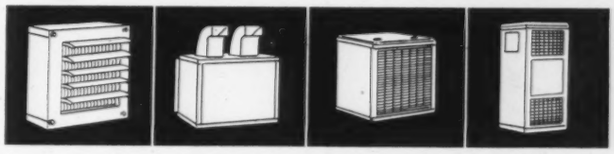
Another UNIT HEATER PROBLEM

Solved by TRANE ENGINEERS

Many of the advancements made in Unit Heater design in the past few years can be attributed to TRANE Engineers. It was TRANE who first introduced the principle of FLOOR LINE SPREAD. That is, getting the heat down into the cold zone where it is needed and wanted.

It was TRANE who first designed a Unit not only applicable to industrial installations, but also for commercial installations in lobbies, stores, auditoriums, and other installations of this character. It was TRANE who introduced the directed heat principle for Blower Type Units and it is TRANE who works so closely with the consulting and heating engineer that TRANE recommendations are accepted as standard by a great many of the leaders in the field.

TRANE UNIT HEATERS



THE MOST COMPLETE LINE MADE

It is also TRANE who first introduced the most complete line of Unit Heaters made. There is a type and size for every application and literature dealing with TRANE Unit Heaters has been simplified to the point where selection of the proper size and type Unit is exceedingly easy.

If you do not have complete TRANE Unit Heater information in your files, send us your request written on your letterhead and complete data will be forwarded to you at once.

Ask Trane first

—About equipment for all Industrial Heating and Air Conditioning installations. TRANE Engineers will be glad to give you every cooperation.

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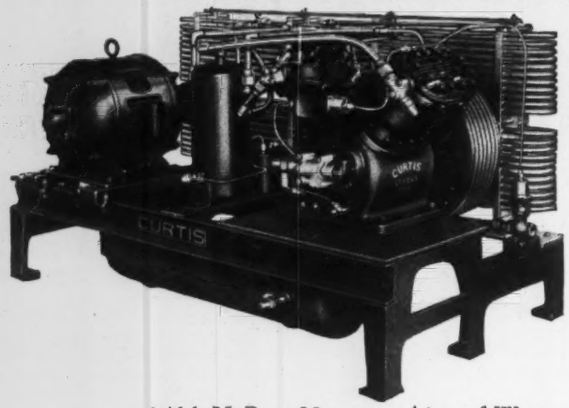
In Canada—Mowat & King Sts. W.—Toronto

CURTIS
AIR-CONDITIONING UNITS AND COMMERCIAL REFRIGERATION



1/6th H. P. to 30 tons—Air and Water cooled. Extra capacity—Slow operating speed—Experienced design—Low up-keep—Rugged construction—Fine materials and Precision workmanship.

Some desirable territories still open.

Advanced design, soundly engineered. 82 successful years—financial stability.

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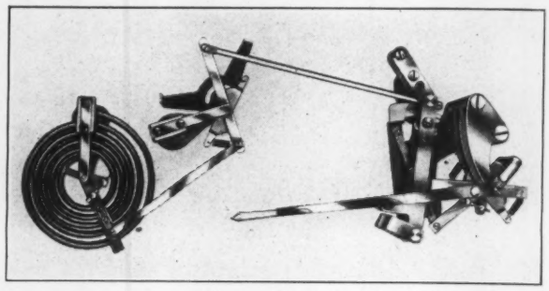
SAME DEPENDABLE PERFORMANCE But-

HIGHER EFFICIENCY and LOWER OPERATING COST

Tested sideby side in our laboratories, this New Vertical Design actually used 25.2% less power than our Horizontal Design—yet the older Horizontal Design matches ANY other make from the standpoint of performance economy. Yes, we have made marked progress toward lower operating costs. Sizes for every vacuum heating requirement. Write for Bulletins 441 and 442A.

AMERICAN STEAM PUMP COMPANY
BATTLE CREEK, MICHIGAN . . . Pumps and Pumps Only Since 1873

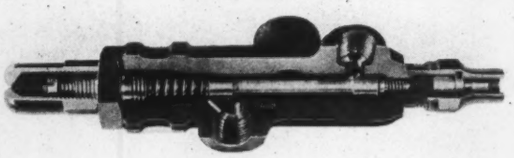
As the line pressure changes, the spiral adjusts the multiplication of the linkage between the float and pen arm. This is not an additive correction but a percentage correction; the linear amount of correction differs at different points of the chart and



is calibrated to fit the flow formula of the meter. The compensator is sensitive to, and will correct, the position of the flow pen for a change in pressure as small as 1/10 of 1 per cent, says the manufacturer.—The Foxboro Co., 106 Neponset Ave., Foxboro, Mass.

Valve Maintains Uniform Oil Flow to Burners

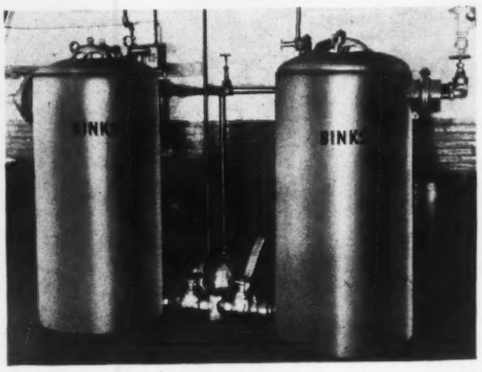
No. 1029—"Uniflow" oil regulating valve has been designed to maintain uniform flow to oil burner despite fluctuations in oil temperature, viscosity, or pressure. It depends for its operation upon the uniform and unvarying delivery of a displacement pump such as an internal gear type. This uniform delivery is taken into the central cylindrical bore of the valve. There are two outlets from this central bore, one leading back to the storage

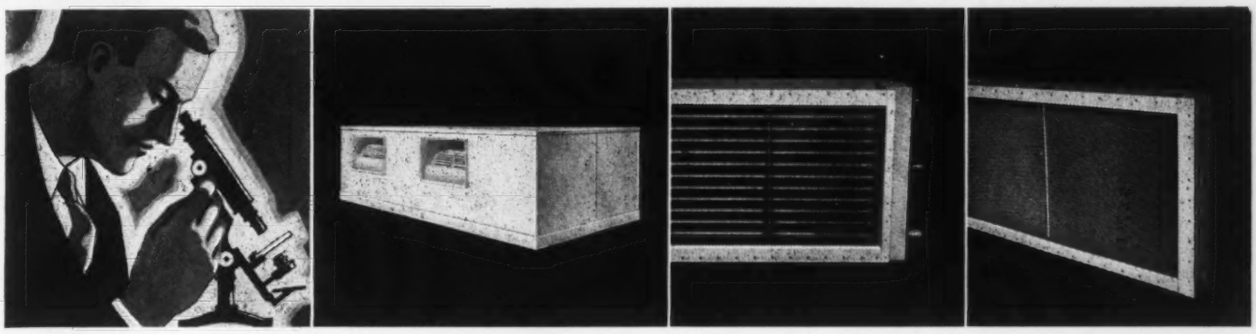


tank, the other to the burner. A piston working back and forth in this bore varies, by means of a screw adjustment, the area exposed by the two outlets of the central chamber. With any position of the piston, therefore, a ratio is determined between the outlet to the tank and the outlet to the burner. The discharge from the pump divides in conformance with this ratio, and, since the discharge is a constant, regardless of the physical state of the oil, the feed to the burner is a constant.—Ace Engineering Co., 1735 W. 31st St., Chicago, Ill.

Sound Deadener Insulation Spray Tanks

No. 1030—Designed especially for spraying sound deadener on car bodies, but said to have a wide variety of uses, including spraying on insulation for air conditioning apparatus, the photo shows an installation of "sound deadening tanks" in an automobile plant. The tanks are usually arranged in pairs in order





What to look for in Air Conditioning Coils and Units

A close study of TRANE Coils and Units will uncover some startling facts. In TRANE "Exclusive Feature" Coils, direct expansion or water, you'll find rugged construction that speaks eloquently for skilled planning and master craftsmanship. In TRANE Cooling and Air Conditioning Units, you'll see simplicity of design, balanced assembly, and a Unit in which each part has been specifically made to operate only with every other part. That is why TRANE Air Conditioning Units have a reputation for being quiet, efficient and economical.

Study the equipment you specify. Compare it with TRANE Equipment and against TRANE Standards. Your final selection of TRANE will pay you dividends in the surety that your installation is well made.

Ask Trane first

—About all industrial heating and air conditioning equipment. A Product is made for every purpose. Unit Heaters, Blast Heaters, Residential Conditioners, Convectors and Heating Specialties.

The TRANE Company has developed a Psychrometric Pad and Ruler which allows air conditioning problems to be calculated with lightening-like speed. We will be glad to let you have the Psychrometric Pad at cost. Simply write us on your letterhead, enclosing thirty cents in coin.

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Jointite TRADE MARK CORK INSULATION FOR AIR CONDITIONING

For economical temperature control, be sure that "Jointite" Corkboard and Moulded Cork Pipe Covering are in the specifications. These products are fabricated by the Mundet process from 100% pure Cork—nature's own insulator—and offer low cost, permanent protection. If you are considering air conditioning, let us show you "Jointite's" performance record . . . its own best reason for being preferred.

Machinery vibration is effectively isolated with Mundet Natural Cork Mats. Ask us for details.

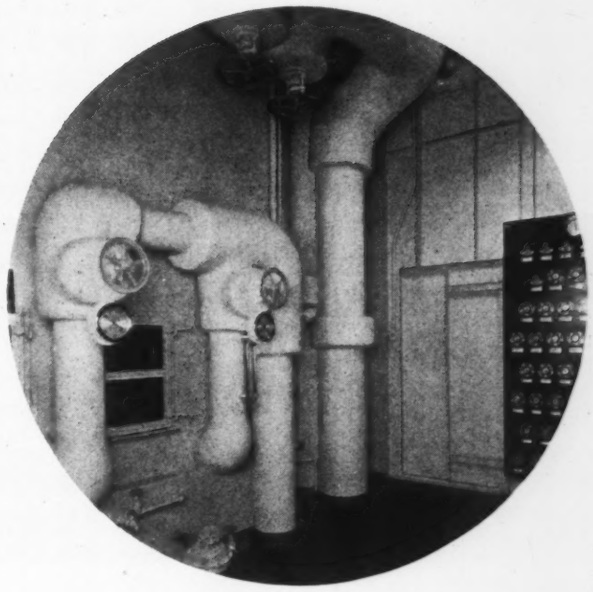

Illustration shows "Jointite" Moulded Cork Pipe Covering with canvas finish.

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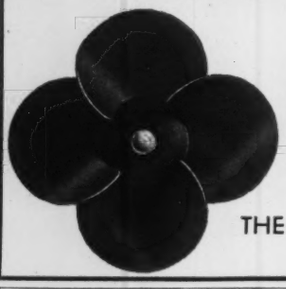
FANS AND BLOWER WHEELS



Patented Alloy Aluminum Blower Wheels. Balanced, Efficient, Rugged, Quiet.

New patented quiet propeller fan blade. The most outstanding development in its field in recent years.

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GLOBE

STEEL INSERTS

Strength
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Economy

● Approved by National Board of Fire Underwriters. Write for catalog showing complete line of installation supplies.

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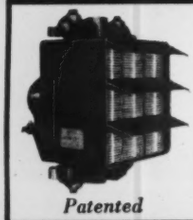
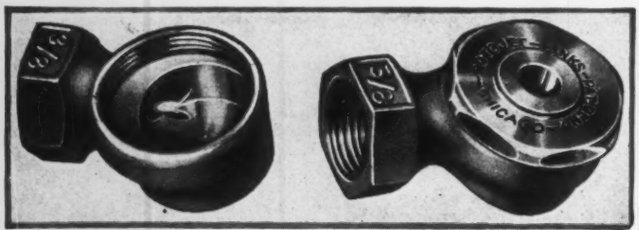
UNIT HEATER



Never fails from corrosion produced by electrolytic action. It is all-cast aluminum and iron radiation—*no other unit heater like it.* Built to last indefinitely without maintenance. Send for details now.

Patented

THE UNIT HEATER & COOLER CO. WAUSAU, WIS.



AIR WASHING NOZZLES

ROTOJET Nozzles furnished in sizes from 1/4" to 2 1/2" pipe connections. Nozzle with cap removed shows involute whirl chamber. Binks ROTOJET Clog-Proof Spray Nozzles are the first choice of engineers everywhere for more efficient AIR WASHING. Send for Bulletin 50 today.

BINKS MANUFACTURING CO., 3106 Carroll Ave., Chicago
WINDSOR, ONTARIO, CANADA

that spraying may be continuous. A 2 stage, 1 hp compressor maintains pressure on the material in the tanks, and a transfer pump is supplied to draw the material from its original container.

The tanks shown each have a capacity of 125 gal and deliver 1½ gpm through approximately 100 ft of 3 in. pipe. The pressure variation in the tanks is from 160 lb to 200 lb and, to maintain this pressure, the compressor operates about one-half of the time.—Binks Mfg. Co., 3106 Carroll Ave., Chicago, Ill.

Recent Trade Literature

For your convenience in obtaining copies of bulletins, see coupon on page 130

No. 2038—AIR CONDITIONING UNITS: The Trane Co., 303 S. 11th St., La Crosse, Wis. 8 p. bulletin on "Climate Changer," a unit for year round air conditioning of small buildings with duct distribution; construction details are described, apartment building applications discussed, and heating and cooling capacities, roughing-in dimensions and suspension suggestions are given.

No. 2039—AIR FILTERS: Wilson & Co., 4100 S. Ashland Ave., Chicago, Ill. 6 p. bulletin describing "Spider-Web" animal hair air filter replacement unit with "Edgeseal" pad, discussing advantages, construction efficiency, and resistance.

No. 2040—ALLOYS: The International Nickel Co., Inc., 67 Wall St., New York, N. Y. 20 p. buyers' guide to warehouse stocks of nickel alloy steels, listing sources of supply by city and state and steels available at each.

No. 2041—BOILERS: Johnston Bros., Inc., Ferrysburg, Mich. (a) 8 p. bulletin giving ratings for steel heating boilers, smokeless type for hand firing, direct draft type for hand firing, mechanically fired type with 3 in. tubes, deep waterleg types with 3 in. and 3½ in. tubes; (b) 8 p. bulletin on firebox boilers, describing and illustrating construction in detail, and giving specifications and dimension sheets for the various types; (c) 8 p. bulletin on Scotch boilers for mechanical firing with coal, gas or oil, describing advantages and construction, and giving dimension sheets.

No. 2042—COMPRESSORS: Worthington Pump & Machinery Corp., Harrison, N. J. (a) 8 p. bulletin on single tandem horizontal two stage steam or motor driven air compressors, describing construction details and giving specifications; (b) 8 p. bulletin on single horizontal three stage steam or motor driven air compressors with similar information.

No. 2043—CONDENSING UNITS: Westinghouse Electric & Mfg. Co., Mansfield, O. Three data sheets on air conditioning condensing units with hermetically sealed casing, giving complete description and specifications for vertical two cylinder, four cylinder and six cylinder single acting direct connected machines.

No. 2044—CONTROLLERS: The Bristol Co., Waterbury, Conn. (a) 8 p. bulletin on "Ampliset Free-Vane" pneumatic controllers for temperature, time-temperature, flow, liquid level, pressure, time-pressure, and humidity with information on applications and construction, as well as accessory equipment; (b) 8 p. bulletin on pneumatic controllers for temperature, liquid level and pressure, explaining principle of operation, and giving data on applications, drilling dimensions, etc.

No. 2045—CONTROLS: Air Conditioning Utilities, Inc., Suite 407-410, 8 W. 40th St., New York, N. Y. Data sheets and bulletins on air conditioning controls, recorders, registers, filters, and seamless flexible metal hose for air conditioning.

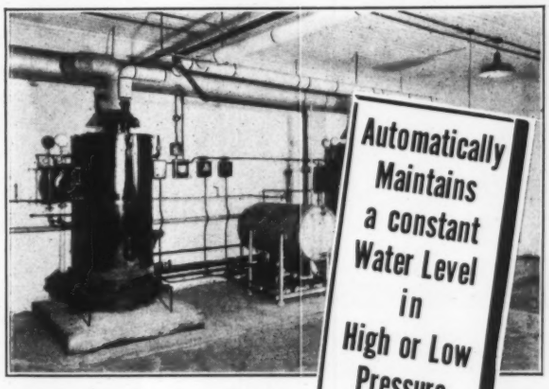
Write for this New Catalog of

MONCRIEF

AIR CONDITIONING PIPE AND FITTINGS

Specially designed and made to meet the exacting requirements of modern air conditioning systems. Moncrief Lock Joints are a valued feature. Everything necessary to install a complete system.

The Henry Furnace & Foundry Co.
3480 E. 49th St. Cleveland, Ohio



Automatically Maintains a constant Water Level in High or Low Pressure BOILERS

Wherever installed the electrically operated M-K-O BOILER FEED has soon paid for itself in increased boiler efficiency and economy.

Entirely automatic in action, this unit pumps feed water against high boiler pressure . . . returns condensate to boiler . . . supplies make-up water as needed. Used with gas, oil, stoker, or hand-fired boilers.

Write today for Bulletin 4-B.

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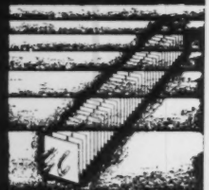

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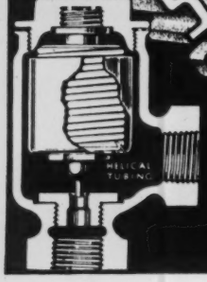

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No. 2046—DUST COLLECTING: Pangborn Corp., Pangborn Blvd., Hagerstown, Md. 16 p. pocket size, ready reference catalog of dust collecting and blast cleaning equipment.

No. 2047—ELECTRIC STEAM GENERATORS: Commonwealth Electric & Mfg. Co., 83-105 Boston St., Boston, Mass. 8 p. bulletin on electric heated steam generators for producing small quantities of steam at high or low pressures (1 to 300 lb gauge), in sizes 2 to 24 kilowatt (1/5 to 2.4 boiler hp), with information on advantages, ratings, construction, pressure control, feed water regulation, etc.

No. 2048—EVAPORATORS: Westinghouse Electric & Mfg. Co., Mansfield, O. Data sheet on extended surface "Freon" evaporator especially designed for comfort air conditioning service, with information on construction, specifications, and performance.

No. 2049—LIQUID LEVEL CONTROLLERS: The McAlear Mfg. Co., 1901-1907 S. Western Ave., Chicago, Ill. 72 p. catalog of liquid level controllers, diaphragm motor valves, lever and float valves, as well as other devices for the control of pressures and levels. The equipment and its construction and application are described in detail, and drawings of applications and roughing-in dimensions are presented. Capacity charts are presented, as are other technical data.

No. 2050—LUBRICATION: The Texas Co., 135 E. 42nd St., New York, N. Y. 16 p. issue of "Lubrication" (Vol. 22, No. 5) devoted to hydraulic power and lubrication of hydraulic equipment, and to low temperature gear lubrication.

No. 2051—PIPE: Taylor Forge & Pipe Works, P. O. Box 485, Chicago, Ill. 4 p. bulletin featuring use of spiral pipe and its light weight advantages.

No. 2052—PIPE MACHINES: Beaver Pipe Tools, Inc., 310 Dana, N. E., Warren, O. 8 p. bulletin on special and standard pipe machines, describing and illustrating them in detail and giving net prices, shipping weights and other information.

No. 2053—PUMPS: Allis-Chalmers Mfg. Co., Milwaukee, Wis. 4 p. leaflet on single suction end inlet centrifugal pumps, direct connected or V-belt drive, bronze fitted, with heads up to 100 ft or over, showing general construction, dimensions and typical combination drives.

No. 2054—PUMPS: Pomona Pump Co., 206 E. Commercial St., Pomona, Cal., and 53 W. Jackson Blvd., Chicago, Ill. 6 p. bulletin "Presenting Pomona's New Hydraulic Test Laboratory," describing facilities for testing efficiency of deep well turbine pumps, and describing the four essential measurements in this type of work and the method of computing pumping costs.

No. 2055—RADIATOR VALVES: American Radiator Co., 40 W. 40th St., New York, N. Y. 4 p. bulletin on air, vacuum and vent valves for venting radiators and mains on one-pipe steam and vacuum systems, featuring new "Variport" and "Hurivent" valves.

No. 2056—REFRIGERATION AND AIR CONDITIONING EQUIPMENT: Acme Industries, Inc., 1700 Cooper St., Jackson, Mich. Loose-leaf catalog presenting information on refrigeration and air conditioning equipment, including many types of pipe coils and bends, headers, nozzles, flanges, fin coils, hangers and brackets, condensers, brine coolers, liquid receivers, separators and traps, tanks, unit coolers, valves and other equipment. Numerous charts and tables of technical data are included.

No. 2057—REFRIGERATION SERVICE: Fedders Mfg. Co., Inc., 57 Tonawanda St., Buffalo, N. Y. 32 p. reprint of articles on refrigeration and engineering service by Joe Askin, Chief Engineer, Fedders Mfg. Co., completely indexed and including a wealth of technical data and information.

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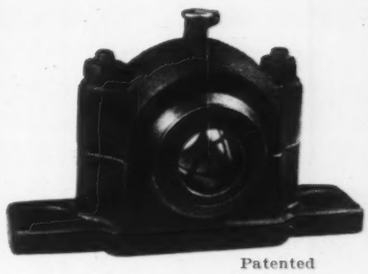

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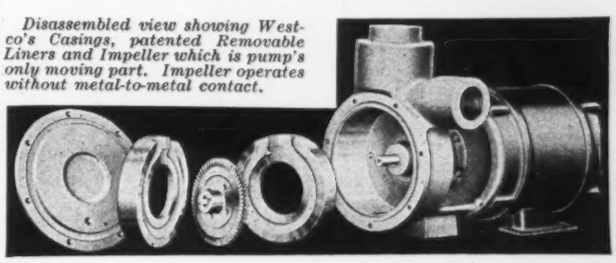

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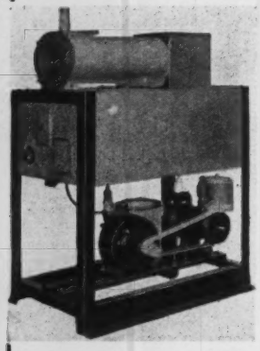

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No. 2059—TAPS: Landis Machine Co., Inc., Tap Division, Waynesboro, Pa. Leaflet on collapsible taps for cutting straight and tapered threads.

No. 2060—VALVES: American Car & Foundry Co., Valve Dept., 30 Church St., New York, N. Y. 64 p. catalog of "a. c. f." full area lubricated plug valves for gasoline, oil, water, air, gas, vacuum, steam, boiler blow-off, chemicals, caustic solutions and other fluids, with information on general characteristics and special features of cylindrical plug type valves, and dimension sheets and drawings for many types, giving directions for installing and servicing, and working pressure ratings and test pressures.

No. 2061—VALVES: Ross Valve Mfg. Co., Inc., Troy, N. Y. (a) 8 p. bulletin on pressure reducing and regulating valves and their applications, with dimensions, sizes and net prices. (b) 16 p. catalog on automatic pressure control valves, feed water filters, grease extractors, float valves and strainers. (c) 8 p. bulletin on altitude and float valves or water level controllers. (d) 12 p. catalog of automatic valves and specialties for water works.

No. 2062—WELDING ELECTRODES: Harnischfeger Corp., 4400 W. National Ave., Milwaukee, Wis. 8 p. booklet on "Smootharc" welding electrodes, describing various types and their applications.

No. 2063—WELDED DESIGN: The Lincoln Electric Co., 12818 Coit Rd., Cleveland, O. Engineering drafting room chart, presenting in ready reference form data necessary for producing arc welded designs, including symbols, information on the 16 types of joints for arc welding, suggestions for design, explanation of nomenclature, and dimensions and properties of base metals and weld metals, and other data. The chart, 24 in. by 35½ in., is printed so as to be suitable for blue printing. Available only to engineers or men in charge of drafting rooms.

No. 2064—X-RAY AND RADIUM INSPECTION: St. John X-Ray Service, Inc., 30-20 Thomson Ave., Long Island City, N. Y. 4 p. folder on the ninety-two elements, with data relating to x-ray work compiled from current literature.

No. 2065—WATER BLENDERS: Sarco Co., Inc., 183 Madison Ave., New York, N. Y. Data sheet on water blenders for automatically mixing hot and cold water, brine or other fluid, and delivering the mixture at any predetermined constant temperature, with description and illustration of unit and typical hook-ups, dimensions, and capacities.

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[7-36]

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